NON-AZEOTROPIC REFRIGERANT MIXTURE (NARM)

VAPOR COMPRESSION CYCLES

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PREFACE

This work investigates the performance of vapor compression cycles with non-azeotropic refrigerant mixtures (NARM's) as the working fluids. Two pure refrigerant and four non-azeotropic refrigerant mixture vapor compression cycle models have been developed. These models were used to evaluate and compare the cycle performance for the temperature range of air conditioner and chiller applications to identify combinations of mixtures and cycles which promise to perform best in these applications.

I wish to express my appreciation and thanks to my thesis adviser Dr. Jeffrey Spitler for his advice to complete this thesis. I also wish to express my gratitude to my advisers, Drs. Jerald Parker and Peter Moretti for their encouragement and advice throughout my graduate program. I also appreciate the helpful comments and advice given by the other members of my committee, Drs. Kenneth Bell and Afshin Ghajar.

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NOMENCLATURE

COP	-	Coefficient of Performance (Cycle only)
COPT	-	Coefficient of Performance (Cle and Condenser Fan)
h	-	Enthalpy
M	-	Mass Flow Rate
P	-	Pressure
PR	-	Compressor Pressure Ratio
s	-	Entropy
T	-	Temperature
UA	-	Heat exchanger Conductance
x	-	Mole Fraction
y	-	Quality

CHAPTER I

INTRODUCTION

1.1 Overview

The concept of a vapor compression refrigeration cycle, sometimes referred to as a reversed Rankine cycle, was initiated and demonstrated in 1834 with ethyl ether (3). The cycle is based on the circulation of a phase-changing working fluid to transport heat from a low temperature heat source to a higher temperature heat sink. The flow diagram of an ideal cycle for a vapor compression cycle is shown in Figure 1. As heat is absorbed isothermally by the evaporator, the working fluid is evaporated at constant pressure and leaves the evaporator as saturated vapor. The saturated vapor enters the compressor and undergoes a reversible adiabatic compression to a temperature (and pressure) level from which heat is rejected in a constant pressure process. The working fluid leaves the condenser as saturated liquid at high pressure and is then flashed to evaporator pressure across an expansion device to complete the cycle.

This process provides significant advantages over other known cooling technologies such as those based on the circulation of working fluids without phase change. Chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) working fluids were invented in the early 1930s by General Motors (42), which was seeking an efficient and safe home refrigerant to replace toxic materials like ammonia and

sulfur dioxide. The CFC's were among the most useful chemical compounds ever developed. This invention provided new motivation to this already established technology and gave rise to an ever-expanding spectrum of useful applications. With CFC as the working fluid, vapor compression cycles are used to refrigerate food and medical supplies, and air condition homes, cars, hospitals, and public buildings etc. In the United States, about 30 percent of the utility power demand is used to drive refrigeration and air-conditioning equipment (50).

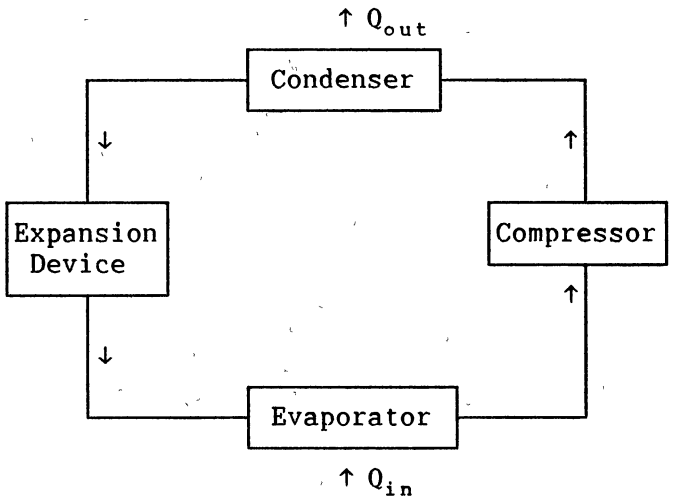


Figure 1. Conventional Vapor Compression Cycle

At the present time, the development of air conditioning and refrigeration services is threatened by external constraints related to environmental protection. During the past decade, researchers have discovered that cholorofluorocarbons migrate to the stratosphere and

chlorine released from CFCs destroys ozone molecules. The stratospheric ozone layer acts as a shield against harmful ultraviolet solar radiation. The direct consequence of the depletion of the ozone layer is that more UV radiation is incident upon the earth's surface, causing a host of problems. Recognizing that the ozone layer depletion is a global problem, the United States and 23 other countries signed the Montreal Protocol to regulate the production and trade of the ozone depleting substances. Fully halogenated CFCs, R-11, R-12, R-113, R-114, and R-115 are covered by the Montreal Protocol as Group 1 (3), (42), and (50).

The elimination of fully halogenated chlorofluorocarbons, which are the basic vapor compression cycle refrigerants used in home appliances, commercial refrigeration, mobile air conditioning, and central air-conditioning systems for large buildings is an additional reason to accelerate research for vapor compression cycles and safer refrigerants.

The air-conditioning and refrigeration industry faces the simultaneous challenges of energy conservation, preservation of the environment, and economic constraints.

Non-azeotropic refrigerant mixtures (NARM's) are a miscible combination of two or more substances; they have been of interest for many years because they offer properties different from those of the limited number of pure refrigerants. NARM's, at saturation, exhibit different mixture compositions in the liquid and vapor phases. The composition of the vapor and the liquid phases varies continuously as the quality changes. This behavior gives NARM's several characteristics which make their use as working fluids in vapor compression cycles

very attractive. The interest in non-azeotropic mixtures of refrigerants for vapor compression systems was stimulated in the 1970s by concern for the availability of fossil energy. Analysis and modeling of refrigeration cycles predicted that significant energy savings are potentially available. This is especially important at the present time when energy and environmental concerns reduce or eliminate from consideration most of the widely used fully-halogenated refrigerants. Although some substitute pure refrigerants have been identified, their availability and commercial acceptance remain to be established.

1.2 Literature Survey

Lorenz and Meutzer (1975) proposed a two-evaporator refrigeration cycle including two additional internal heat exchangers. They were the first to use intercooling to further enhance energy efficiency. From the experimental results, they reported energy savings of 20% using 0.50 R-11/0.50 R-12 mixture compared with pure R-12.

Stoecker (1978) developed a simulation model for a two-evaporator-with-two-intercooler domestic refrigerator and found an improvement of 12% in the COP with 0.50 R-12/0.50 R-114 mixture compared with pure R-12. However, these improvements were not confirmed by experimental measurements.

Dhar and Gadhi (1980) performed a theoretical analysis of a window air conditioner using non-azeotropic refrigerant mixture R-12/R-13. The performance was evaluated for a simple vapor compression cycle. They reported that the use of 0.20 R-13/0.80 R-12 mixture in the existing air-conditioner gives a marginally (3.5%) lower capacity than that with

pure R-22. However, the compressor power was reduced by more than 18% compare with pure R-22 refrigerant system.

Stoecker and Walukas (1981) simulated single and two evaporator refrigerators with R-12/R-114 mixtures. The objective of the simulation studies was to compare the performance of the refrigeration system using a single refrigerant with one using a mixture, both experimentally and analytically. The prime reason for choosing R-12/R-114 were that both R-12 and R-114 are non-polar compounds and form an ideal mixture. Therefore, Raoult and Dalton's laws can be applied for vapor-liquid equilibrium calculations and the coefficient of performance of R-12 and R-114 is about the same when operating under identical conditions.

The simulation was ideal in nature such that saturated liquid left the condenser while saturated vapor entered the compressor. The temperatures of the source fluids entering and leaving the evaporator and condenser were specified. With the requirement of constant cooling load in the evaporator, mass flow rate of the source fluid in evaporator was fixed while that of the sink side in the condenser varied to account for the different amount of condenser heat discharged. All heat exchangers were specified by the product of an overall heat transfer coefficient and area (UA). The unknown state variables resulting from the steady state simulation were solved by employing the Newton-Raphson iteration method. The simulation indicated that at 0.5 mass fraction of R-114, the power requirement is 11.4% less than the mixture at 0.01 mass fraction of R-114. Because the computer program was designed for mixtures only, it could not simulate the case for pure components. In the simulation of two-evaporator refrigerators,

two thirds of the refrigeration load was assumed to be at the low temperature evaporator while the remainder was at high temperature evaporator. The results showed a power saving of 12% at 0.5 mass fraction of R-114. The major drawback of the simulation was the inability to investigate various pure and mixed refrigerants. This study, however, was the first of its kind and demonstrated a general procedure for the simulation of steady state thermal systems.

Kruse (1981) has done experimental work to simulate a NARM vapor compression heat pump. He reported a maximum 25% improvement in coefficient of performance for heating of the 0.50 R-22/0.50 R-114 mixtures in comparison to pure R-22. However, 0.60 R-12/0.40 R-114 mixture exhibited only 15% improvement in coefficient of performance for heating compared with pure R-12.

Stocker (1984) has developed a simulation model to determine energy saving through the use of refrigerant mixtures for heat pump and refrigerator application. A simulation model was developed for a two-evaporator, two-intercooler refrigerator. The simulation predicted an 8% saving in compressor power for 0.32 R-12/0.68 R-114 mixture in comparison to pure R-12 as the working fluid. However, the experimental results showed only 2% saving in compressor power for the R-12/R-114 mixture in the range between 0.85 to 0.90 R-12. They showed the lower condensing heat transfer contributed to the discrepancy between simulation and experiment.

Kruse et al. (1985) improved the previous simulation model by applying the RKS equation of state for vapour-liquid-equilibrium in the prediction of thermodynamic properties of mixtures. The simulation model was used to predict the performance of a heat pump operating with

R-22/R-114 mixtures. The simulation model predicted the heat pump operated with 0.70 R-22/0.3 R-114 mixture had an increase of COP of about 3% when compared to pure R-22. An experimental set up for simulation of tap-water heat pumps was built to compare the prediction and experiment data. With 0.70 R-22/0.30 R-114, the experiment showed an increase in COP of 7% but a decrease in heat output of about 15% when compared to pure R-22. The simulation model with the RKS equation of state was also used to calculate the performance for a two-evaporator refrigerator with R-22/R-114, and R-13B1/R-114 mixtures as working fluids. The model predicted for 0.40 R-22/0.60 R-114 mixture an improvement of 18% in coefficient of performance when compared with pure R-12. With 0.70 R-13B1/0.30 R-114 mixture an improvement 20% in COP when compared with pure R-12 was predicted. The corresponding temperature difference in the evaporator was 18°K and 27.5°K for R-22/R-114 and R-13B1/R-114 mixtures respectively. But the high temperature evaporator area was increased by 35% and 58% for R-22/R-114 and R-13B1/R-114 mixtures respectively.

Djourshari and Radermacher (1986) evaluated the performance of two vapor compression heat pump cycles, one with a single stage solution circuit and one with two stage solution circuit using a mixture of R-22 with another candidate refrigerant in order to propose a safe and nontoxic mixture. It is indicated that both cycles show a significant increase in COP (up to 58% for the two stage cycle) compared to pure R-22. The two stage cycle shows a pressure ratio which is only 45% of the one for pure R-22, resulting in increased mass flow rate but reduced capacity.

Didion and Mulroy (1986) reported the results of laboratory measurements of the change in performance of a substantially unmodified residential heat pump designed for R-22 when charged with a NARM of R-113B1/R-152A. The measurement of performance for various sizes of fixed expansion devices showed that the effect of gliding temperature in the saturation zone is small and the effect of composition shift by flash distillation in the accumulator improves low temperature heating performance.

Cloud et al. (1986) showed that a NARM, coupled with appropriate modification to the basic vapor compression refrigeration cycle, can save up to 20% of the electric power required for the best single refrigerant system. The mixture is also shown to require less heat sink flow - approximately 14 percent for the example given of a long-range patrol aircraft.

Quast and Kruse (1986) have done experimental work with a reciprocating compressor to analyze the compressor behavior with various blends of R-22/R-114. Measurements were made with an open type reciprocating compressor run by a variable speed DC electric motor. The refrigerant mass flow rate, calculated by an energy balance at the heat exchangers, the shaft power and the P-V diagram were measured to compare the influence of different mixture concentrations on the isentropic efficiencies. The result of the experiment indicates that the efficiency decreases slightly with decreasing R-22 concentrations. At R-22 concentrations of less than 40% this effect is greater than at higher R-22 concentrations.

McLinden and Radermacher (1987) developed a program and presented methods for comparing the performance of pure and mixed refrigerants in

the vapor compression cycle. Even though it was designed for heat pump applications, it still can be used for calculating and comparing refrigeration COP and capacity. All thermodynamic properties needed for the simulation were computed by Carnahan-Starling-DeSantis (CSD) equation of state developed by Morrison and McLinden. The program called CYCLE7 is capable of simulating the ideal vapor compression cycle for pure and mixed refrigerants.

Radermacher and Lavell (1988) used CYCLE7 to compare the performance of an R-22/R-142B mixture against that of R-12. The results showed increases in both COP and pressure ratio of up to 12 and 16% respectively and decrease in capacity of 8% for an optimized mixture.

Mulroy et. al (1988) measured a 32% efficiency improvement on an experimental, water-to-water, breadboard heat pump apparatus when the non-azeotropic mixture of R-22/R-114 replaced R-22.

Parent and Larue (1989) developed a simulation model for a water-to-air heat pump operating with a non-azeotropic refrigerant mixture. The mixture properties were based on CSD equation of state and performance was evaluated for a conventional cycle. This model has been validated with experimental results for three non-azeotropic mixtures R-13B1/R-152a, R-22/R-114, and R-23/R-22. These experiments were used to check the accuracy of a general model using non-azeotropic mixtures. Mixtures were chosen to validate the simulation, rather than to optimize performance. The results indicated the model showed an accuracy of around 5% to 6% on thermodynamic behavior for all refrigerants tested.

Troxel and Braven (1989) developed a computer model to estimate the performance of a liquid-liquid heat pump system using nonazeotropic

refrigerant mixtures as the working fluid. The NARM properties were based on the Carnahan-Starling-DeSantis (CSD) equation of state. To examine the potential of these mixtures and associated equipment constraints, only the basic vapor compression cycle was considered. R-11/R-22, R-114/R-22 and R-152a/R-22 mixtures were used in this study. They reported for the R-114/R-22 mixtures, the composition in the range of 0.30 to 0.60 mole fraction R-114, the coefficient of performance for the mixture working fluid was 18% increased over a system using a working fluid of pure R-22.

Vineyard (1989) used 0.60 R-22/0.40 R-142 mixture (mass fraction) for a domestic refrigerator designed for pure R-12 without any modifications. He reported that refrigerator with R-22/R-142b mixture required 8.6% more energy than for pure R-12 system.

Grzyll and Silvestri (1990) discussed metering problems, caused by the thermostatic expansion valves (TXV). These valves are spring preloaded to provide a constant amount of superheat at the evaporator outlet. Until now, TXV bulbs are usually charged with some standard refrigerant such as R-11,R-12,R-22 etc. If the NARM heat pump uses a standard TXV, system performance is extremely unpredictable. It is possible that the system will not operate at all. A computer model has been used to simulate the "true" effects of adding refrigerant mixtures to heat pump systems containing common TXV's. Using R-22 as the base line system, the performance of systems using mixtures of R-115/R-22, R-13B1/R-22,R-12/R-22, and R-290/R-22 was estimated. The result indicated that only R-13B1/R-22 could operate over the entire concentration range, with a decrease in system capacity. R-115 and R-12 mixture can operate only over specific concentration ranges and with a

decrease in system capacity. Mixtures of R-22 with either R-12 or R-11 can not operate in any concentration.

Stanger et al. (1990) using a computer model to investigate the effects of constant heat exchanger area on the coefficient of performance for liquid-liquid heat pumps analyzed systems which use nonazeotropic mixtures. Two computer models were developed to estimate the heat exchanger area based on two methods. In the first method, log mean temperature differences (LMTDs) through the heat exchangers were specified, and were held constant for all refrigerant compositions. In the second method the heat exchanger UA product was kept constant, thus approximating constant heat exchanger area over a range of refrigerant compositions. Two NARM's, R-22/R-11, and R-22/R-114 were used as the working fluids, and the results indicated only one percent difference in COP prediction between the two methods.

Ro et al. (1990) used the Peng-Robinson equation of state to estimate the thermodynamic properties of the refrigerant mixtures. Their heat pump model was based on conventional vapor compression cycle, and two types of simulations (modulation of heating capacity with a constant volumetric flow rate and performance enhancement with a constant heating capacity) were carried out to characterize the performance of the heat pump using R-22/R-152a, R-22/R-142b, R-22/R-114 and R-13b1/R-152a as working fluids. They reported that it was possible to modulate heating capacity in a heat pump with a displacement type compressor and to increase coefficient of performance in case of a constant heating capacity.

Smith et al. (1990) used NIST's thermodynamic property subroutine to develop a simulation model for a two-evaporator refrigerator with

two intercoolers. The simulation were used to locate the limitation of NARM system. Three NARM's, R-22/R-142b, R-22/R-123, R-32/R-142b were studied. The results indicated that an ideal NARM system (constant air temperature) had a limiting coefficient of performance that was less than that of the best performing pure refrigerant component. With a non-ideal NARM system (gliding air temperature), the NARM-based had a higher limiting coefficient of performance than a system running on either pure constituent of the NARM. The effects of the intercoolers to the COP of NARM system were significant.

Jung (1990) created a simulation model to predict the performance for a conventional domestic refrigerator, with the refrigerant thermodynamic properties based on CSD equation of state. The objectives of the study were identify the best substitutes for R-12. The results indicated that only R-22/R-142b and R-32/R-142b mixtures yielded an increase in COP (up to 3%) with the same capacity as that of R-12.

Mokadam et al. (1991) created a simulation model for a NARM advanced vapor compression cycle. The cycle operated with a flooded evaporator and a two-phase outlet condenser. The NIST's thermodynamic properties program was used for simulation. Mixtures R-113/R-22 and NH3/H2O were simulated and they reported that the system operated with these NARM's could get a high coefficient of performance with a low compressor pressure ratio. However the comparison of performance with pure refrigerant system was not reported.

The summary of the literature survey is given in Table I.

The above review indicates that most of the studies of NARM vapor compression cycles concentrated on the refrigerator and heat pump applications and the performance of system usually evaluated on the

TABLE I

LITERATURE SURVEY SUMMARY

Author	Year	Ref.	Applications	Working fluids	System	E.O.S	Experi.	Model
Lorenz	1975	25	Refrigerator	R11/R12	Two-evaporator		X	
Stoecker	1978	54	Refrigerator	R12/R114	Two-evaporator			X
Dhar	1980	9	Air conditioner	R13/R12	Conventional			X
Stoecker	1981	55	Refrigerator	R12/R14	Conven. & two-evap.	Ideal	X	
Kruse	1981	20	Heat pump	R22/R114	Conventional		X	
Stoecker	1984	56	Refrigerator	R12/R14	Two-evaporator	Ideal	X	
Kruse	1985	21	Heat pump	R22/R114	Conventional	RKS	X	X
Djourshari	1986	11	Heat pump	R22/DEGDME	1,2 Stage Solution		X	
Didion	1986	10	Heat pump	R113B1/R152a	Conventional		X	
Cloud	1986	6	Cooling		2-stage compressor			X
Quast	1986	44	Refrigerator	R22/R114	Conventional	RKS		
McLinden	1987	28			Conventional			
Radermacher	1988	46		R22/R142b				X
Mulroy	1988	41	Heat pump	R22/R114			X	
Parent	1989	43	Heat pump	R13B1/R152a,R22/R114 R23/R22		CSD	X	X
Troxel	1989	57	Heat pump	R11/R22,R114/R22 R152a/R22	Conventional	CSD		X
Grzyll	1990	18	Heat pump	R115/R22,R13B1/R22 R12/R22,R290/R22	Conventional	CSD		X
Stanger	1990	53	Heat pump	R22/R11,R22/R114	Conventional	CSD		X
Ro	1990	47	Heat pump	R22/R152a,R22/R142b R22/R114,R13B1/R152a	Conventional	PR		
Smith	1990	48	Refrigeration	R22/R142b,R22/R123, R32/R142b	Two-evaporator	CSD		X
Jung	1990	19	Refrigeration	R22/R142b,R32/R142b	Conventional	CSD		X
Mokadam	1990	35	Cooling/heating	R113/R22,NH3/H20	2-phase eva outlet	CSD		X

conventional vapor cycle suitable for pure refrigerants. There was very little work done on the performance of advanced cycles for air conditioner or chiller application. Further, most of the previously investigated NARM's contained at least one refrigerant which belongs to group 1 of the Montreal Protocol. The literature survey indicates a lack of research in the area of NARM vapor compression cycles and safe working fluids to take advantage of non-azeotropic refrigerant mixture properties in the chiller and air conditioner applications.

1.3 Objectives

In the United States, the National Appliance Energy Conservation Act (NAECA) was enacted in 1987, which established energy efficiency standards for several consumer appliances including refrigerator-freezers. The NAECA requires the Department of Energy to determine acceptable standards for refrigerator-freezers, starting January 1,1990. At the same time, the Environmental Protection Agency (EPA) issued a proposed rule to freeze production of R-11, R-12, R-113, and R-115 at 1986 levels due to their relative ozone depletion potential. This freeze will be followed in mid-1993 by a 20 percent reduction from 1986 levels, and in mid-1998 by a 50 percent reduction from 1986 levels (8).

The air-conditioning and refrigeration industry is faced with the problem of trying to reduce power consumption in the same time period in which it must begin using replacement refrigerants that may increase energy consumption. Adding to this dilemma, most changes in refrigeration system design require long lead times to implement in production.

Demand KW of conventional HVAC equipment for an average commercial building are distributed as follows :

Refrigeration equipment :	
Compressor	58 %
Cooling tower fan	8 %
Air side system :	
Total fan	27 %
Pumping system :	
Chiller water	3 %
Condenser water	4 %

Two main items, the refrigerant compressor and the air side fans consume 85 percent of the total electric power. Increasing the performance of vapor compression cycle can reduce the power consumption and demand KW of these components and has a great effect on the economics of the cooling system.

The application of non-azeotropic refrigerant mixtures to vapor compression cycles can reduce the power required by these systems. A NARM differs from a single refrigerant in that, for a constant pressure evaporation and condensation, the NARM saturation temperature varies, causing by the composition change of the liquid and vapor phases which occur during the phase change process. This is the so-called 'gliding temperature interval', while the temperature of a single substance remains constant during a constant pressure phase change process.

The change in temperature during the phase change process can be shown to have thermodynamic benefits in terms of higher coefficient of performance COP in a refrigeration cycle because of:

- smaller mean temperature differences between heat absorption and heat rejection.

- a reduction in irreversible heat transfer losses owing to lower mean temperature gradients in the evaporator and condenser (between the refrigerant and heat source or sink fluid), and

- because the refrigerant mixture can be tailored to a given application to minimize pressure ratio and component size. In addition, the temperature change in the condenser allows a large change in the heat sink temperature, which substantially reduces heat sink flow rate requirements for a given heat load. This reduces the power draw for the condenser fan in system using air as the heat sink.

The application of NARM vapor compression cycles can be improved further with the development of advanced cycles with features such as two-phase refrigerant at the exit of the evaporator and condenser. By changing these conditions (two phase refrigerant qualities), the pressure ratio of the compressing process can be reduced and the COP can be increased.

With the application of non-azeotropic refrigerant mixtures for vapor cycle systems, another variable is added, mixture composition. To gain the most benefit from NARM, the selection of the NARM and the applied cycle must be taken into consideration in analysis and design of systems.

The design of an energy efficient vapor cycle system requires consideration of a spectrum of system design variables. These include selection of the refrigerant, the arrangement of components (thermodynamic cycle), the cycle states and the component design. All of these variables have a direct impact on the system performance

(power and heat sink demands) and system weight (weight is important for aircraft application). For commercial or industrial applications, the benefit criterion would most likely be defined in direct economic terms such as cost of ownership. Its components would include acquisition and operating costs with the factors of size (weight) and efficiency.

Other important factors include safety of refrigerant, power draw, heat exchanger sizes as well as complexity which will have an impact on cost and reliability.

In considering the interrelationship between these variables, it is useful to examine the thermodynamic characteristics of the refrigeration cycle.

For the vapor compression refrigeration cycle, to get maximum COP, the temperature at which heat is absorbed by the refrigerant in the evaporator should approach the heat source temperature and the temperature at which heat is rejected by the refrigerant in the condenser should approach that of the heat sink fluid; i.e., the cycle temperature ratio should be minimized.

The thermodynamically desirable objective for small temperature differences across the evaporator and condenser has to be traded with the resulting heat exchanger sizes which increase with decreasing temperature difference. In the case of the condenser, the temperature difference across the heat sink fluid itself will be affected by the refrigerant-to-sink ΔT. If the heat sink is air, clearly the condenser air flow rate will be affected. Within the cycle itself, there are other design variables which influence weight and performance. The

larger (more effective) the subcoolers, the higher the system COP but the higher the weight/cost.

The refrigeration cycle, the working fluid, and the states at which the refrigerant operates in the cycle have to be selected carefully in order to obtain an optimum system from the user standpoint.

Over the past several decades, there has been a considerable amount of research in the use of refrigerant mixtures in vapor cycle systems. However, most of the studies of NARM vapor compression cycles concentrated on refrigerator and heat pump applications and the performance of system usually evaluated on the conventional vapor cycle suitable for pure refrigerants. There was very little work done on the performance of advanced cycles for air conditioner or chiller application. Further, most of the previously investigated NARM's contained at least one refrigerant from group 1 of the Montreal Protocol. (Group 1 includes R-11, R-12, R-113, R-114 and R-115)

The above review indicates a lack of safe working fluids and vapor compression cycles that take advantage of NARM properties in the area of chiller and air conditioner, and therefore, research needs to be continued to identify an arrangement of the components and appropriate refrigerant mixtures to improve the performance of the vapor cycle and to identify blends which have acceptably low ozone depletion potential.

The objectives of this study are:

1. To examine and evaluate alternative cycles designed to take advantages of unique NARM properties.
2. To develop methodology to select safe non-azeotropic refrigerant mixtures for various cycles.

3. To develop a simulation model to predict the performance of various vapor compression cycles using NARM as a working fluid.
4. To identify the most promising combination of non-azeotropic refrigerant mixtures and cycles for various operating conditions in the temperature ranges of air conditioner and chiller application.

The cycles considered for this study are given in Figure 2. and they are:

- Cycle 1, Figure 2a. This cycle is a conventional vapor compression cycle and used as a base case to compare the performance with other cycles. The working fluids are pure refrigerants or azeotropic refrigerant mixtures. The system consists of a condenser, an evaporator, a single stage compressor and an expansion valve. Saturated or slightly superheated vapor exits the evaporator and the saturated or subcooled liquid exits the condenser.

- Cycle 2, Figure 2b. The system consists of a condenser, an evaporator, a 2-stage compressor, a subcooler, and two expansion devices. Saturated or slightly superheated vapor exits the evaporator and saturated or subcooled liquid exits the condenser. The working fluids are pure refrigerants or azeotropic refrigerant mixtures.

- Cycle 3, Figure 2c. The system is similar to the Cycle 1, except the system operates with a non-azeotropic refrigerant mixture.

- Cycle 4, Figure 2d. The system consists of a condenser, an evaporator, a 2-stage compressor, two intercoolers, and three expansion devices. Saturated or slightly superheated vapor exits the evaporator

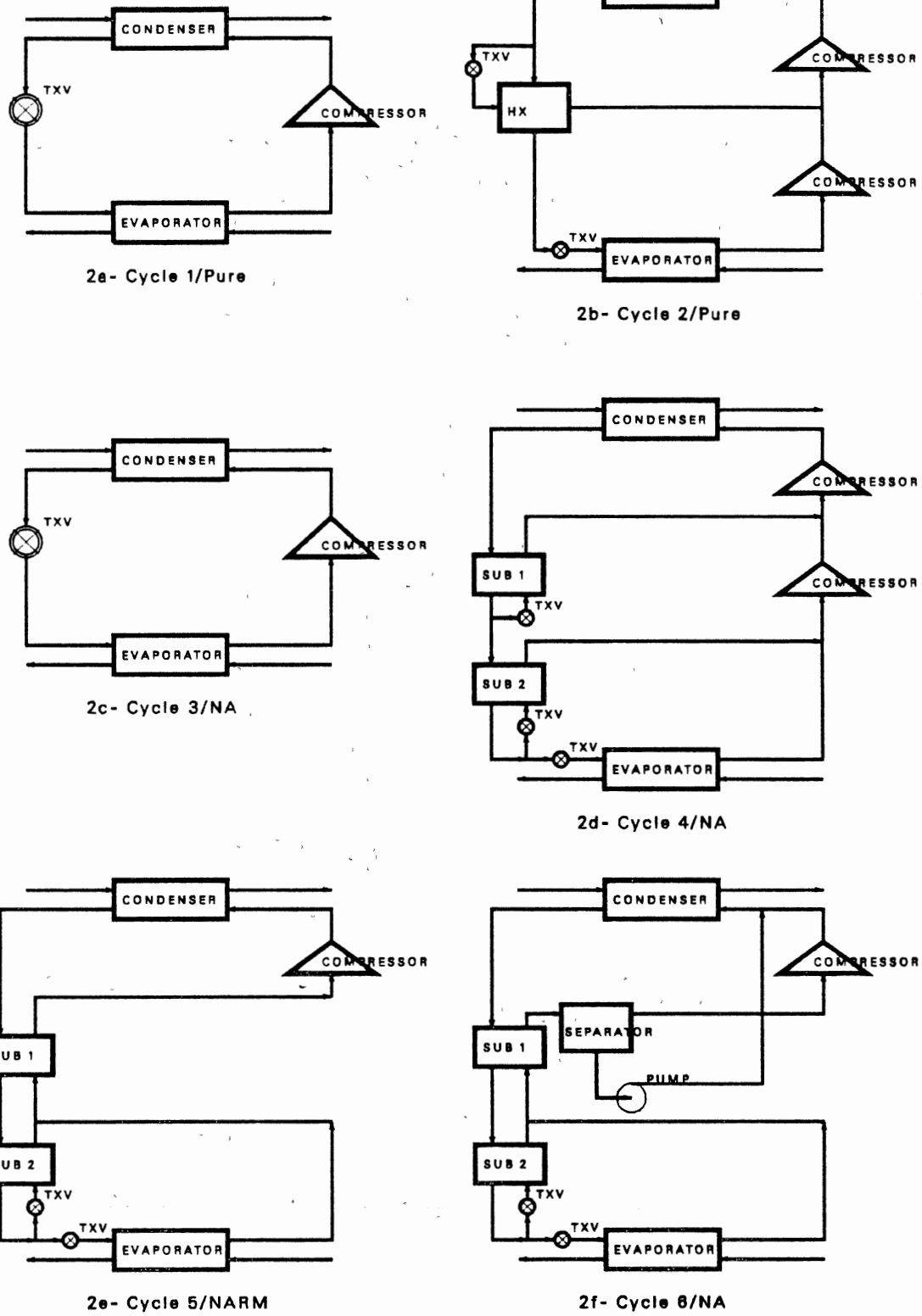


Figure 2. The Cycles Considered in this Study

and saturated or subcooled liquid exits the condenser. A NARM is the working fluid.

- Cycle 5, Figure 2e. The system consists of a condenser, an evaporator, a single stage compressor, two subcoolers, and two expansion devices. Low pressure two-phase refrigerant exits the evaporator and the high pressure two-phase refrigerant exits the condenser. A NARM is the working fluid.

- Cycle 6, Figure 2f. The system consists of a condenser, an evaporator, a single stage compressor, three subcoolers, and two expansion devices, a separator and a liquid pump. Low pressure two-phase refrigerant exits the evaporator and high pressure two-phase refrigerant exits the condenser. A NARM is the working fluid.

The constituents of the NARM's used for this study are selected the from the safe refrigerants of National Institute of Standards and Technology database (16).

The NIST Thermodynamic Properties of Refrigerants and Refrigerant Mixtures Database (version 2.0) is used to calculate the cycle performance (16).

Performances are evaluated for application in the operating temperature ranges of chillers and air-conditioning.

1.4 Terminology

Heat source side: Hot side fluid in the evaporator.

Heat sink side: Cold side fluid in the condenser.

NARM: Non-Azeotropic Refrigerant Mixture.

CHAPTER II

DESCRIPTION OF NARM PROPERTIES AND NARM CYCLES

Mixtures of refrigerants offer special characteristics that are not possessed by the individual pure refrigerant. Vapor compression systems with refrigerant mixtures as the working fluid can help reduce the power consumption in refrigeration and air conditioning. At the same time, it can help relieve global climatic change in two ways : decreased electrical consumption compared to conventional compression vapor cycle systems (hence decreased power plant emissions and greenhouse gases), and using refrigerant mixtures, we can increase the choice of working fluid candidates for the vapor compression cycle to operate with safer refrigerants which have acceptably low ozone depletion potential.

In this chapter we review the thermodynamic properties of refrigerant mixtures, describe the equations of state (EOS) used to predict the properties of refrigerant mixtures, consider the behavior of non-azeotropic refrigerant mixtures in the vapor compression cycle, and describe the cycles considered in this study. The mixtures are limited to binary mixtures.

2.1 Properties of Refrigerant Mixtures

The references for section 2.1 are from (1),(31),(49) and (59). The phase rule for a nonreacting PVT system is described by:

$$F = 2 - P + n \qquad (2.1)$$

Where F : number of degrees of freedom of the mixture

P : phase existing in the mixture

n : number of chemical species in the mixture

For a nonreacting PVT system containing two chemical species (n=2), the phase rule F = 2-P+n becomes F=4-P, and the maximum number of independent intensive variables required to specify the thermodynamic state of the stable system is therefore three, corresponding to the case of a single phase (P=1). If the three intensive variables are chosen as pressure (P), temperature (T), and one of composition (x), then all equilibrium states of the system can be represented in three-dimensional P-T-x space. Within this space, the state of pairs of phases, define a surface. Similarly, the state of three phases in equilibrium is represented as a curve. Two-dimensional phase diagrams, such as temperature-composition at a constant pressure (T-x phase diagram), are useful to describe phase equilibrium conditions of a binary mixture. Two-dimensional phase diagrams are obtained from intersections of the three-dimensional surfaces and planes of constant pressure or constant temperature. The curve in a two-dimensional phase diagram represents the states of saturated liquid mixtures; it is called the boiling or "bubble-point" curve. Similar, the condensing or "dew-point" curve, represents states of saturated vapor. In a T-x phase diagram of a binary mixture, an example of which is shown in Figure 3, the region above the dew-point curve is superheated vapor and the region below the bubble point curve is subcooled liquid. The region between the bubble point curve and the dew point curve is the two-phase zone. In a P-x phase diagram of a binary

mixture, an example of which is shown in Figure 4, the region below the dew-point curve is superheated vapor and the region above the bubble point curve is subcooled liquid. The region between the bubble point curve and the dew point curve is the two-phase zone.

Ideally, when two refrigerants A and B are combined in a A-B binary solution, the mixture properties will show a proportional relationship with the concentration x of the mixture. For example the vapor pressure of each component in a liquid solution for instance, called the partial pressures P_A, P_B, are proportional to the corresponding mixture concentration for an ideal mixture (Raoult's law).

Application of Raoult's law to each component gives:

$$x_A Psat_A = y_A P \qquad (2.2)$$

$$x_B Psat_B = y_B P \qquad (2.3)$$

The mole fractions in each phase sum to unity

$$x_A + x_B = 1. \qquad (2.4)$$

$$y_A + y_B = 1. \qquad (2.5)$$

where x_A,x_B : mole fraction of components A,B respectively in the mixture A-B liquid.

y_A,y_B : mole fraction of components A,B respectively in the mixture A-B vapor.

$Psat_A$,$Psat_B$: saturation pressure of pure components A,B respectively at the same temperature T.

If the total pressure is not very high, Dalton's law will also be obeyed and the total pressure will equal the sum of the partial pressures.

In the case of the Dalton model, the properties of each component

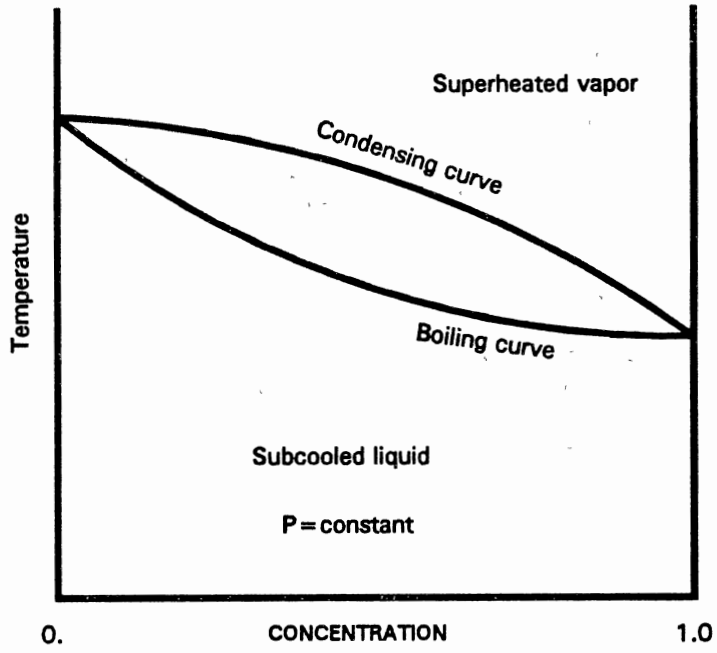


Figure 3. T-X Diagram

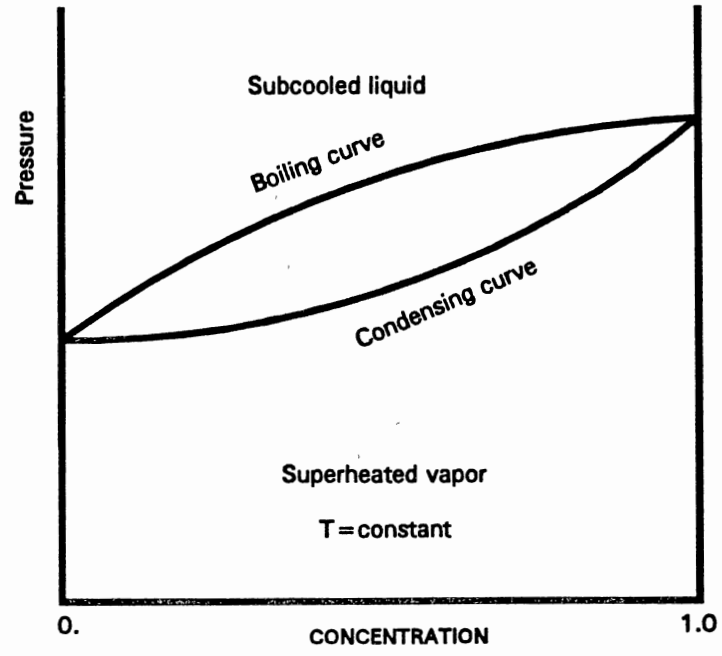


Figure 4. P-X Diagram

are considered as though each component existed separately at the volume and temperature of the mixture. Consider this model for the special case in which both the binary mixture A-B and the separated components A,B can be considered as ideal gases.

For the mixture : $PV = nRT$ (2.6)

$$n = n_A + n_B \quad (2.7)$$

For the component : $P_A V = n_A RT$ (2.8)

$$P_B V = n_B RT \quad (2.9)$$

On substituting we have

$$P = P_A + P_B \quad (2.10)$$

The P-x phase diagram of an ideal binary mixture following Raoult's law is given in Figure 5a which shows that the bubble-point curve of a system which obeys Raoult's law is linear. Similarly, the partial pressures $P_A = y_A P$, $P_B = y_B P$ of the two components are proportional to the x_A, x_B composition, respectively, as shown by the two dashed lines. This figure gives the composition of the liquid phase corresponding to a given vapor pressure. The T-x phase diagram is shown in Figure 5b.

In general, however, the composition of the vapor in equilibrium with the liquid will be richer in the more volatile component than will be the composition in the liquid phase. Some special cases, such as mixtures for which the bubble-point and partial pressure curves lie above the Raoult's law lines as in Figure 6a, are said to exhibit positive deviations from Raoult's law. Similarly, if these curves fall below the Raoult's law lines as show in Figure 7a, the mixtures show negative deviations from Raoult's law. When the deviation from Raoult's law becomes very large (this may be a result of polarity differences), causing the total vapor pressure curve to pass through a maximum or

minimum. Significant deviation from Raoult's law is often manifested by azeotrope formation at a composition called azeotrope composition.

The bubble-point curve and T-x phase diagram of a binary solution with a positive deviation from Raoult's law that passes through a minimum boiling-point azeotrope are given in Figures 6a and 6b.

The bubble-point curve and T-x phase diagram of a binary solution with a negative deviation from Raoult's law that passes through a maximum boiling-point azeotrope are given in Figures 7a and 7b.

The highest possible vapor pressure in a mixture occurs as the sum of the individual vapor pressures at a particular temperature when two liquid phases coexist, and which are completely immiscible with each other. A mixture with immiscible liquids in a certain concentration range is shown in Figure 8.

The binary refrigerant mixture types can be selected as fluids for vapor compression cycles as follows:

- Azeotropic mixture: A mixture of two or more different fluids is called "Azeotropic" if the composition of its components in the vapor is the same as that of the liquid in the equilibrium condition at a given pressure.

The azeotropic state of a binary system is special in that it possesses only a single degree of freedom, rather than two as required for normal two-component, two-phase equilibrium. Thus specification of any one of the coordinates T,P or composition for a binary azeotropic mixture fixes the other two, provided that the azeotrope actually exists. Binary azeotropic mixtures are therefore similar to the saturation states of pure components. The transition of an azeotropic mixture from liquid to vapor (or vice-versa) occurs at constant

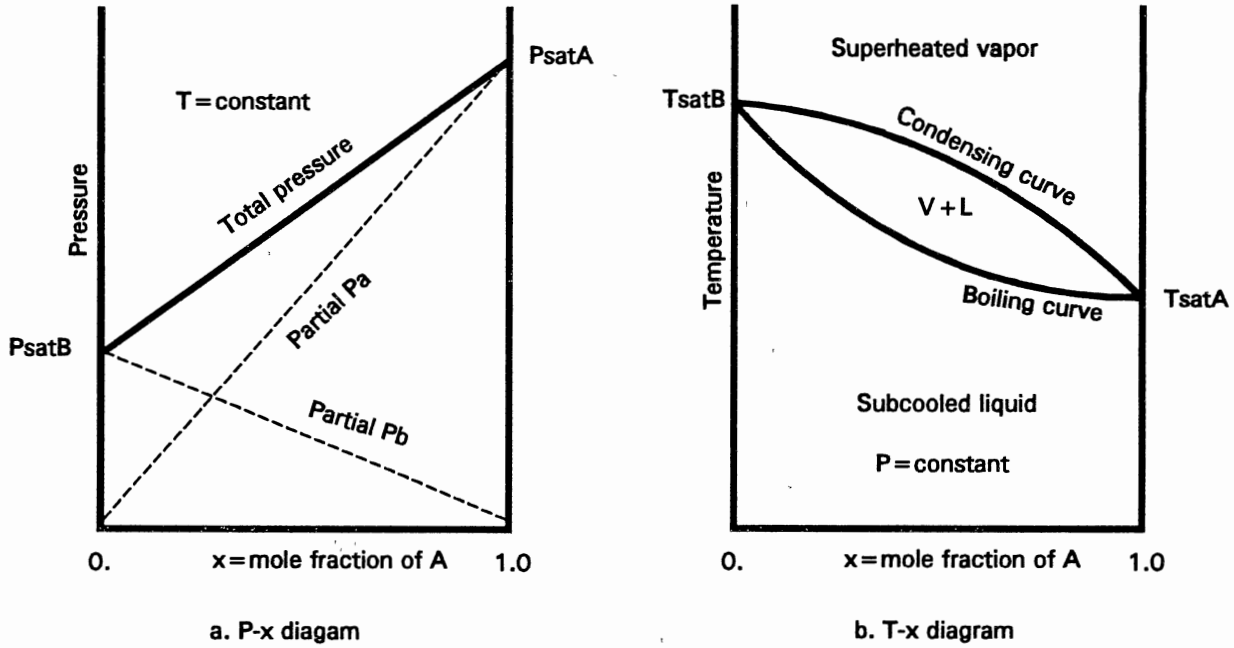


Figure 5. Mixture Follows Dalton's Rule

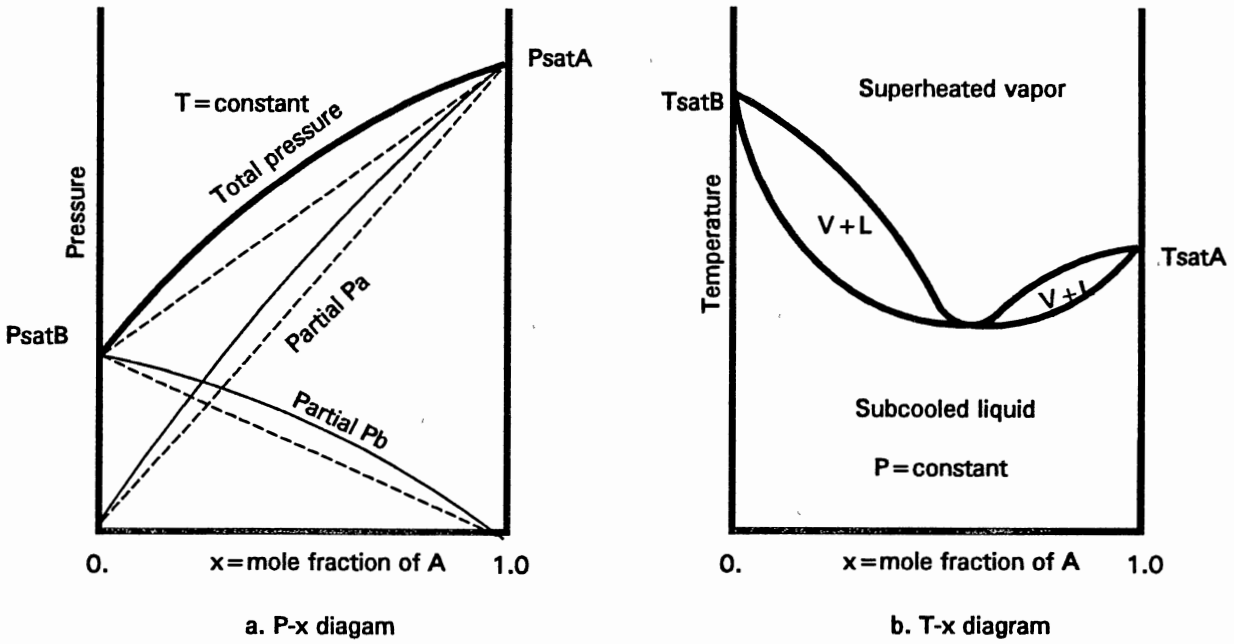


Figure 6. Mixture with a Minimum Boiling-Point Azeotrope

+re deviation from Raoult's law

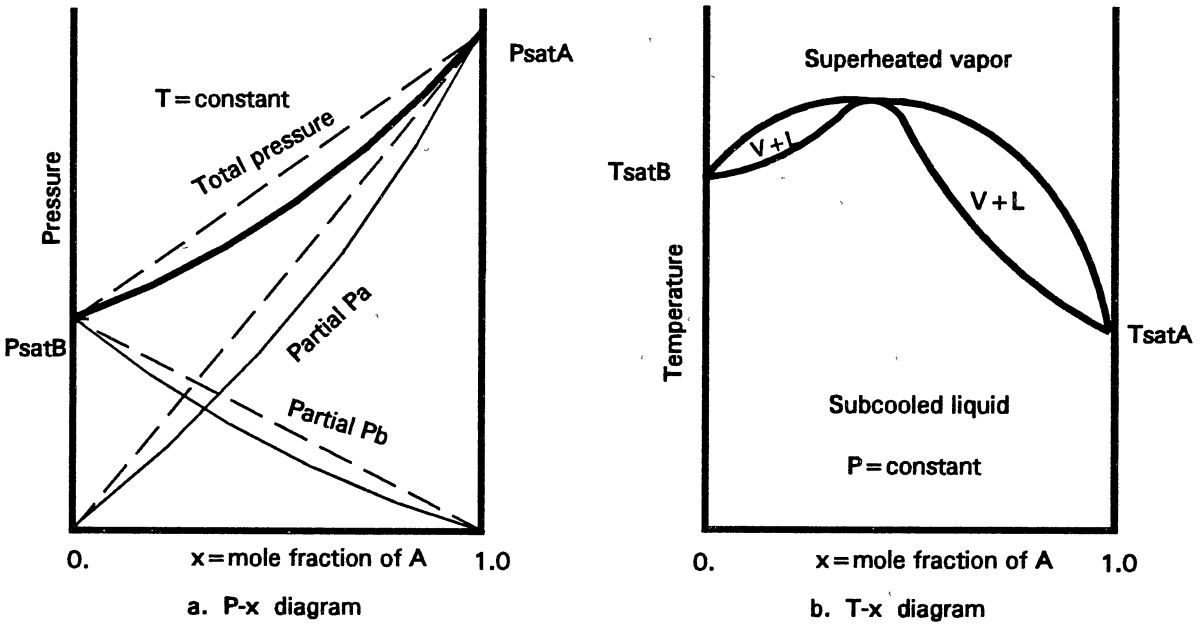


Figure 7. Mixture with a Maximum Boiling-Point Azeotrope

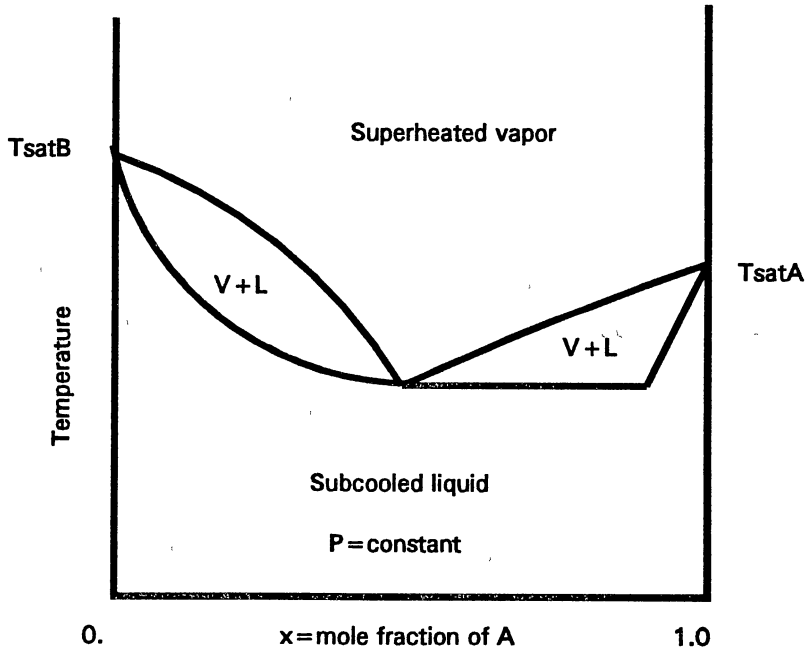


Figure 8. T-X Diagram of Limited Solubility Mixture

temperature and pressure without any change in composition.

An important characteristic of the azeotropic state is the occurrence of a minimum or a maximum on the phase diagram at the azeotropic composition on both bubble point and dew point curves. As mentioned earlier, the behavior of an azeotropic refrigerant mixture in boiling and condensing is similar to an individual pure refrigerant. But they may be used for different boiling point requirements or to advantageously modify the properties of pure refrigerants. The most successful commercially available azeotropes to date in the refrigeration industry are the R-500 series which contain one fully halogenated and one partly halogenated CFC such as R-500(R-12/R-152a), R-501(R-22/R-12), R-502(R-22/R-115), R-503(R-23/R-13), R-504(R-32/R-115), R-505(R-12/R-31) and R-506(R-31/R-114).

Non azeotropic mixtures: Mixtures that do not form an azeotrope at any concentration, ie., mixtures whose total pressure curve does not pass through an minimum or maximum value, are defined as "non-azeotropic" mixtures.

To understand the behavior of a non-azeotropic binary mixture we use an imaginary experiment and a temperature versus composition diagram. Figure 9 shows a piston-cylinder arrangement containing a non-azeotropic binary liquid mixture A-B with a composition of constituent A, x_1. The piston has a fixed mass and is frictionless so that the pressure of the mixture is always constant. Figure 10 is the T-x diagram for the mixture with the initial subcooled liquid mixture state shown as point 1. As heat is slowly transferred to the mixture, it reaches the bubble point curve at point 2, and vapor will begin to form and collect under the piston as shown in Figure 9a. If the experiment

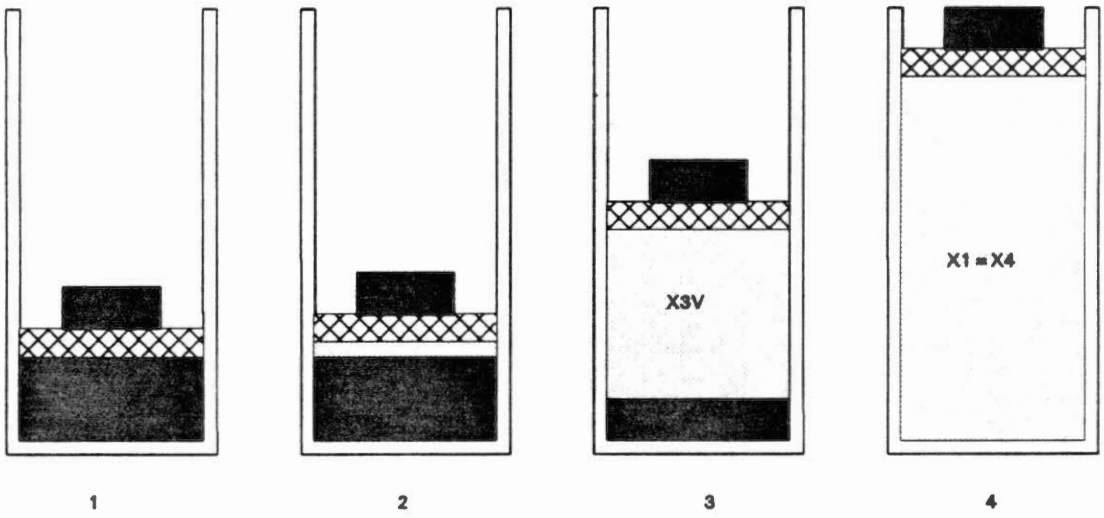


Figure 9. Experiment of NARM Behavior

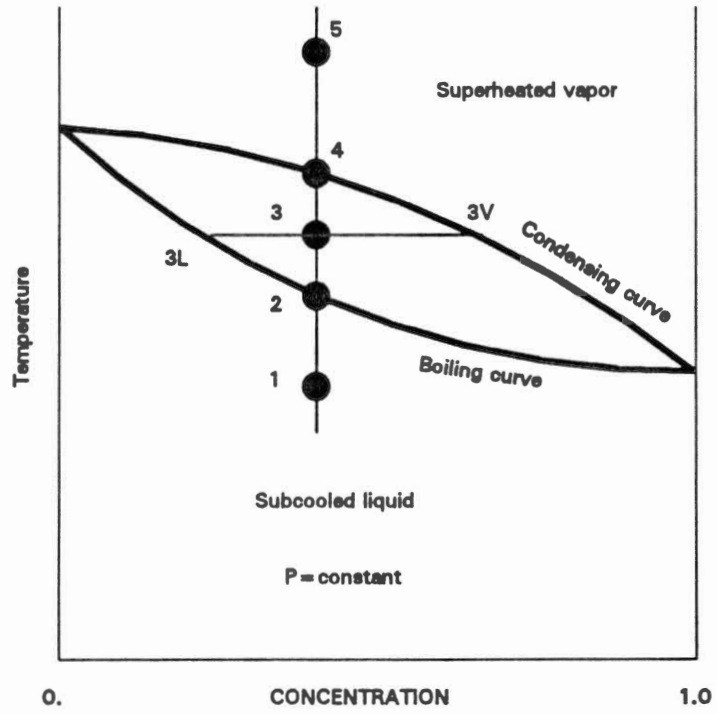


Figure 10. T-X Diagram of NARM

is stopped at some point above this temperature, such as at state point 3 we find the composition of the constituent A in the liquid might be at point 3L while the composition of the constituent A in the vapor is at point 3V. The composition of constituent A in the vapor x_{3V} is larger than composition of constituent A in the liquid phase x_{3L}. As more heat is added, the remaining liquid phase becomes richer in the less volatile component B which raises the boiling point of the remaining liquid. Continued heating of the mixture will gradually vaporize all of the liquid to saturated vapor at state point 4, at point 4, $x_4=x_1$. Further heating will superheat the vapor to point 5. When the superheated vapor mixture is cooled at constant pressure, the entire process will be reversed. Additional experiments carried out at different compositions at the same pressure will establish the bubble-point curve and dew point curve. Unlike a pure substance, non-azeotropic mixtures do not have a single saturation temperature for each pressure, because the saturation temperature is a function of the composition of the mixture.

2.2 Equation of State

Equation of state are used for three major purposes:

- Prediction of the equilibrium values in phase behavior calculations.

- Calculation of the pressure-volume-temperature relationships, density and other physical properties of mixtures.

- Determination of the values of enthalpy and entropy.

The simplest of all equations is the perfect gas law,

$$Z = PV/RT = 1. \qquad (2.11)$$

It is appropriate only for gases at very low pressure (less than 15. psia). At a higher pressure, the virial equation of state is used

$$Z = 1. + B(T)/V + C(T)/V^2 + \ldots . \qquad (2.12)$$

When this equation is truncated at the second virial coefficient, B(T), it is appropriate to 150. psia in the vapor phase.

Although the virial equation of state can in principle be used to fit gas-phase isotherms to any required accuracy, experimentally determined virial coefficients beyond the third are scarce because very extensive and precise PVT data are required for their evaluation. Unfortunately, simultaneous description of both the liquid and vapor portions of subcritical isotherms cannot be accomplished with a truncated virial equation of state and a single set of virial coefficients. Therefore, prediction and correlation of the volumetric properties of fluids at high densities and in the liquid region are usually done with empirical equations of state.

All PVT equations of state are empirical equations. The coefficients and the arrangement of variables in equations were found based on a particular data set available to the investigator.

Equations of state are well-known in the literature. The equations are simply repeated here for the completeness of this study.

P : absolute pressure

T : absolute temperature

v : molar volume

a : temperature and composition-dependent parameter

b : composition-dependent parameter

subscript i,j for each component

2.2.1 Peng-Robinson Equation of State

The references for P-R equation of state are from (39),(47)

$$P = \frac{RT}{v-b} - \frac{a(T)}{v(v+b)+b(v-b)} \tag{2.13}$$

For each pure component, i, the parameters are

$$a_i(T) = a_{ci}(T_{ci})\alpha_i(T) \tag{2.14}$$

$$a_{ci}(T_{ci}) = 0.45724 \frac{(R.T_{ci})^2}{P_{ci}} \tag{2.15}$$

As usual, $\alpha_i(T)$ is expressed as follows

$$\alpha_i(T) = 1 + (1 - \frac{T}{T_{ci}})(m + n \frac{T}{T_{ci}}) \tag{2.16}$$

$$b_i = 0.07780 \frac{R.T_{ci}}{P_{ci}} \tag{2.17}$$

By using the compressibility factor Z, a cubic equation of the following form is obtained.

$$Z^3 - (1-b)Z^2 + (A-3B^2-2B)Z - (AB-B^2-B^3) = 0. \tag{2.18}$$

$$\text{Where } Z = \frac{Pv}{RT}; \quad A = \frac{aP}{R^2T^2}; \quad B = \frac{bP}{RT} \tag{2.19}$$

Two parameters, m and n in $\alpha_i(T)$ can be determined from the experimental data for saturated states. Experimental data for vapor pressure can be taken from ASHRAE's vapor pressure relation for pure refrigerants and the fugacity equality can be used to find m and n.

The equation of state for mixtures can be completed by assigning the proper mixing parameters for the components. The conventional mixing rules can be applied.

$$a = \Sigma\Sigma_{ij} (x_i x_j a_{ij}) \tag{2.20}$$

$$b = \Sigma\Sigma_{ij} (x_i x_j b_{ij}) \tag{2.21}$$

$$a_{ij} = (1-k_{ij})(a_i a_j)^{.5} \tag{2.22}$$

$$b_{ij} = (1-kk_{ij})(b_i+b_j)/2. \tag{2.23}$$

The formula for enthalpy, h and fugacity coefficient are:

$$\ln(\Phi_i) = \frac{b_i}{b}(Z-1) - \ln(Z-B) - \frac{A[(2\Sigma\Sigma_{ij}(x_j a_{ii})/a - b_i/b]Z^*}{2.828\sqrt{B}} \tag{2.24}$$

$$\frac{\Delta H}{RT} = (Z-1) + \frac{[T(da/dt)-a]Z^*}{2.828\sqrt{B}} \tag{2.25}$$

$$\text{Where } Z^* = \ln[(Z+2.414B)/(Z-.414B)] \tag{2.26}$$

2.2.2 Redlich-Kwong-Soave EOS

The references for RKS EOS are from (22),(39), and (63)

Soave version of the Redlich-Kwong EOS is as follows:

$$P = \frac{RT}{v-b} - \frac{a(T)}{v(v+b)} \tag{2.27}$$

For each pure component, i, the parameters are

$$a_i(T) = a_{ci}(T_{ci})\alpha_i(T) \tag{2.28}$$

$$a_{ci}(T_{ci}) = 0.42748\frac{(R.T_{ci})^2}{P_{ci}} \tag{2.29}$$

As usual, $\alpha_i(T)$ is expressed in (2.16)

$$b_i = 0.08664\frac{R.T_{ci}}{P_{ci}} \tag{2.30}$$

By using the compressibility factor Z, a cubic equation of the following form is obtained.

$$Z^3 - Z^2 + (A-B-B_2)Z - AB = 0. \tag{2.31}$$

$$\text{Where } Z = \frac{Pv}{RT}; \quad A = \frac{aP}{R^2T^2}; \quad B = \frac{bP}{RT} \tag{2.32}$$

Two parameters, m and n in $\alpha_i(T)$ and mixing parameters can be determined by the same method in 2.2.1.

The formula fugacity coefficient are:

$$\ln(\Phi_i) = \frac{b_i}{b}(Z-1) - \ln(Z-B) - \frac{A}{B}[(2\Sigma\Sigma_{ij}(x_j a_{ij})/a - b_i/b]Z^* \quad (2.33)$$

Where $Z^* = \ln(1+B/Z)$ (2.34)

2.2.3 Carnahan-Starling-DeSantis EOS (CSD)

The CSD equation of state has been shown by Morrison and McLinden (1985) to be capable describe both the liquid and the vapor properties, including those under the vapor dome, of refrigerants as well as refrigerant mixtures, with a single equation of state. Carnahan and Starling (1969) proposed the form and DeSantis et al. (1976) used the same form to describe the phase behavior of mixtures of simple fluids. This modification, referred to as the Carnahan-Starling-DeSantis (CSD) equation of state (15),(16),36),(37):

$$\frac{Pv}{RT} = \frac{1 + y + y^2 - y^3}{(1-y)^3} - \frac{a(T)}{RT[v+b(T)]} \quad (2.35)$$

Where $y = b/4v$. The first term on the right side of equation is the Carnahan-Starling representation of the behavior of a system of rigid, noninteracting spherical molecules; the parameter b is related to this hard-sphere

volume and proposed by DeSantis et al.

For each pure component, i, the parameters are

$$a_i(T) = a_{0i}\ EXP(a_{1i}T + a_{2i}\ T^2) \quad (2.36)$$

$$b_i(T) = b_{0i} + b_{1i}T + b_{2i}T^2 \quad (2.37)$$

The coefficients a_{0i}, a_{1i}, a_{2i}, b_{0i}, b_{1i}, b_{2i} are estimated based on empirical data for a given refrigerant.

2.3 NARM Behavior in a Vapor Compression Cycle

To understand the behavior of a NARM in a vapor compression cycle, we use as a base line a conventional vapor compression refrigeration system having an evaporator, a condenser, a compressor and an expansion device. The compressor receives low pressure and temperature refrigerant from the evaporator and compresses it to a high pressure. This compression process is associated with an increase of refrigerant temperature. The high temperature and pressure vapor enters the condenser. The refrigerant, passing through the condenser, rejects heat to a high temperature heat sink and condenses to a subcooled liquid. Then, the refrigerant flows through the expansion device undergoing a drop in pressure and temperature. Finally, the low pressure, low temperature, and low quality refrigerant enters the evaporator, where it picks up heat from the low temperature heat source, reaching a superheated (or high quality) vapor at the evaporator exit. When this cycle is applied in a air conditioning system, the low temperature heat source is the indoor environment and the high temperature heat sink are the outdoor environment.

This base line cycle helps to explain the reason for using a non-azeotropic refrigerant mixture in a vapor compression cycle. The refrigerant absorbs heat in the evaporator at low temperature, and after compression, rejects heat in the condenser at high temperature. If the refrigerant is a pure substance, the temperature of the refrigerant will remain constant in the evaporator as long as two-phase

refrigerant is present. The temperature of the heat source fluid will, however, decrease in temperature over the flow length of the evaporator, as shown in Figure 11a. Similarly, in the condenser, the temperature of the refrigerant will remain constant in the presence of liquid and vapor phase refrigerant while the temperature of the heat sink fluid will increase over the flow length of the condenser as shown in Figure 12a. The interest in using nonazeotropic mixtures of refrigerants stems from the fact that constant pressure evaporation or condensation of a NARM occurs with a change in temperature. When counterflow evaporators or condensers are used in a system with a NARM, the temperature profile will be more parallel than they would be in a conventional system . Using parallel temperature profiles lowers the thermodynamic irreversibility of the system.

Reversible heat transfer occurs if the temperature difference between the two fluids is zero and the surface area approaches infinity. Large temperature differences result in large irreversibilities. A second law analysis of these heat transfer processes suggests that the irreversible energy losses can be minimized by maintaining a constant temperature difference between the two heat transfer fluids over the flow lengths of the condenser and evaporator. If the temperature difference between the heat transfer fluids is held constant, the mean temperature difference required to transfer a given amount of heat will be less than the mean temperature difference required for the same duty using single component refrigerants. Thus, by using NARM's, the mean evaporation temperature can be increased, and the mean condensation temperature can be decreased. From theoretical considerations, it can readily be shown that an increase in evaporation

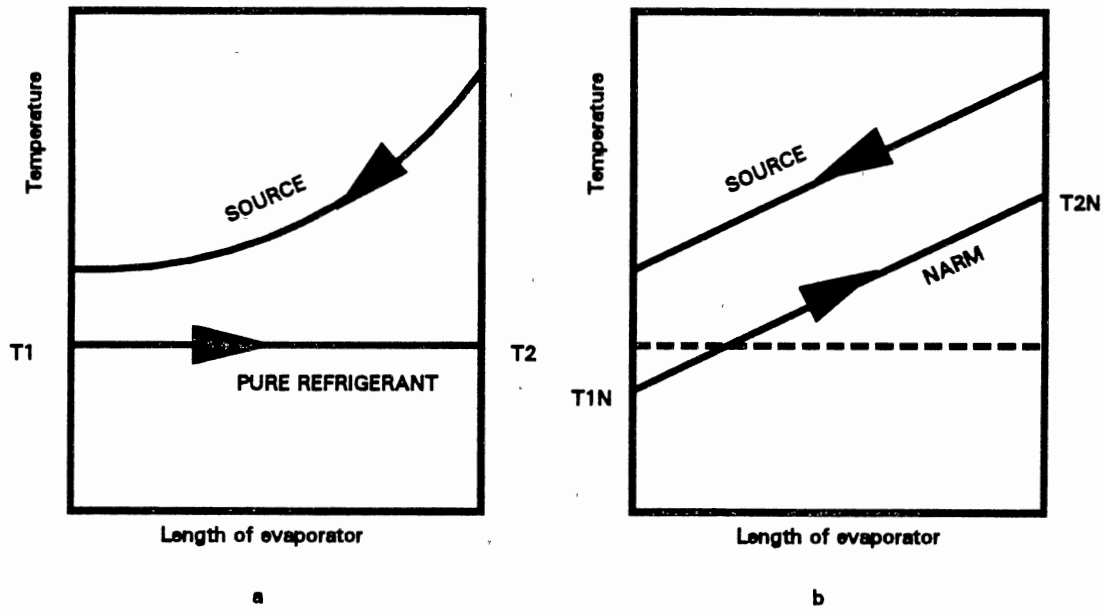


Figure 11. Temperature Profile in a Counterflow Evaporator

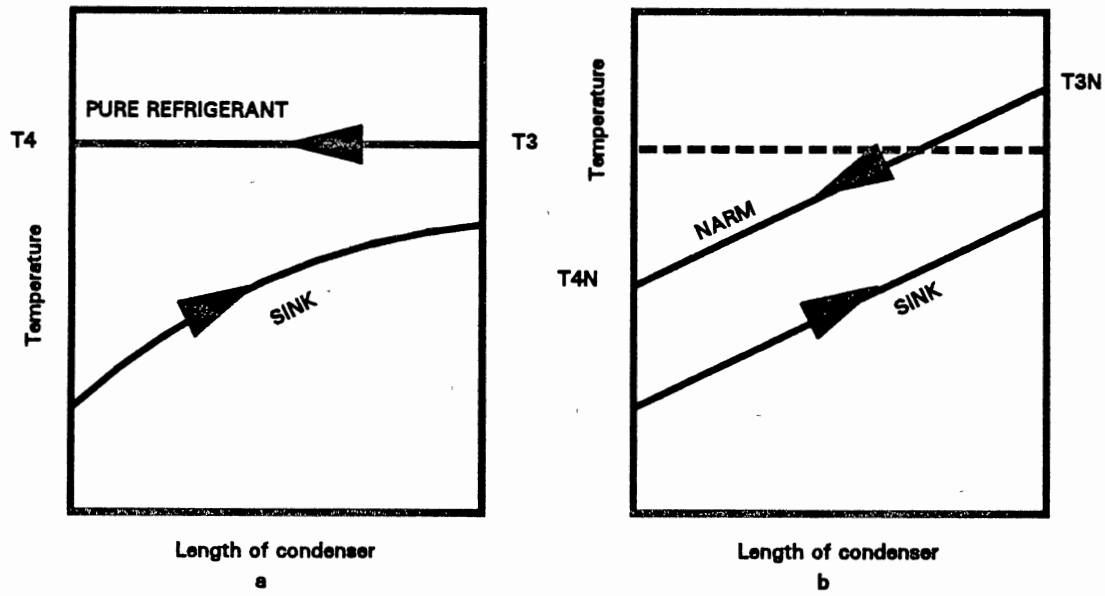


Figure 12. Temperature Profile in a Counterflow Condenser

temperature and a decrease in condensation temperature will yield an increase in the coefficient of performance. To illustrate the above idea, Figures 11b and 12b show a comparison between refrigerants when the required refrigeration duty is the same. The source fluid enters and exits at the same temperatures in both cases. But in the case of the pure refrigerant, there is a large difference between the inlet temperature of source fluid and the exit temperature of the evaporating refrigerant. The condensing process is similar to the evaporating process. The sink fluid enters and exits at the same temperatures in both cases. But in the case of pure refrigerant, there is a large difference between the inlet temperature of sink fluid and the exit temperature of the condensing refrigerant.

Through choice of refrigerant composition and mass flow rates, it is theoretically possible to maintain a constant temperature difference between the refrigerant and the second fluid over the flow length of a counterflow heat exchanger. Assuming the evaporators for both pure refrigerant and NARM have the same area A, heat transfer rate Q_{evap}, and over all heat transfer coefficient U, this constant temperature difference for NARM is given by:

$$\Delta T_{NARM} = T_{so\ in} - T_{N\ out} = T_{so\ out} - T_{N\ in} \qquad (2.38)$$

Where $T_{so\ in}$: source inlet temperature

$T_{so\ out}$: source outlet temperature

$T_{N\ in}$: NARM inlet temperature

$T_{N\ out}$: NARM outlet temperature

for pure refrigerant, let $\Delta T_1 = T_{so\ in} - T_{p\ out}$ (2.39)

$$\Delta T_2 = T_{so\ out} - T_{p\ in} \tag{2.40}$$

where $T_{p\ in}$: pure refrigerant inlet temperature

$T_{p\ out}$: pure refrigerant outlet temperature

and $T_{p\ in} = T_{p\ out}$ (2.41)

$$\Delta T_{pure} = \frac{(\Delta T_1 - \Delta T_2)}{\ln [\Delta T_1 / \Delta T_2]} \tag{2.42}$$

In order for the refrigeration duties in the two cases to be identical, the following must be true:

$$\Delta T_{pure} = \Delta T_{NARM} \tag{2.43}$$

It is easy to prove that

$$\frac{\Delta T_1 + \Delta T_2}{2} > \frac{\Delta T_1 - \Delta T_2}{\ln [\Delta T_1 / \Delta T_2]} = \Delta T_{NARM} \tag{2.44}$$

The inequality 2.44 show that the mean temperature of the source and NARM (left side of Inequality 2.44) is less than the average temperature difference between the source and pure refrigerant. In other words, the mean temperature of NARM evaporation is higher than the evaporation temperature for a pure refrigerant. The left side of Inequality 2.44 is equal the right side only if $\Delta T_1 = \Delta T_2$ (the refrigerant and the source have constant temperature difference).

Similarly, for the condenser, we can prove the mean temperature of NARM condensation is lower than the condensation temperature for a pure refrigerant. If we look at both refrigeration cycles on a temperature-entropy diagram (assuming isentropic compression and expansion in both cases) and include the condensation processes, we can compare efficiencies by comparing the areas in Figure 13. The cycle for the

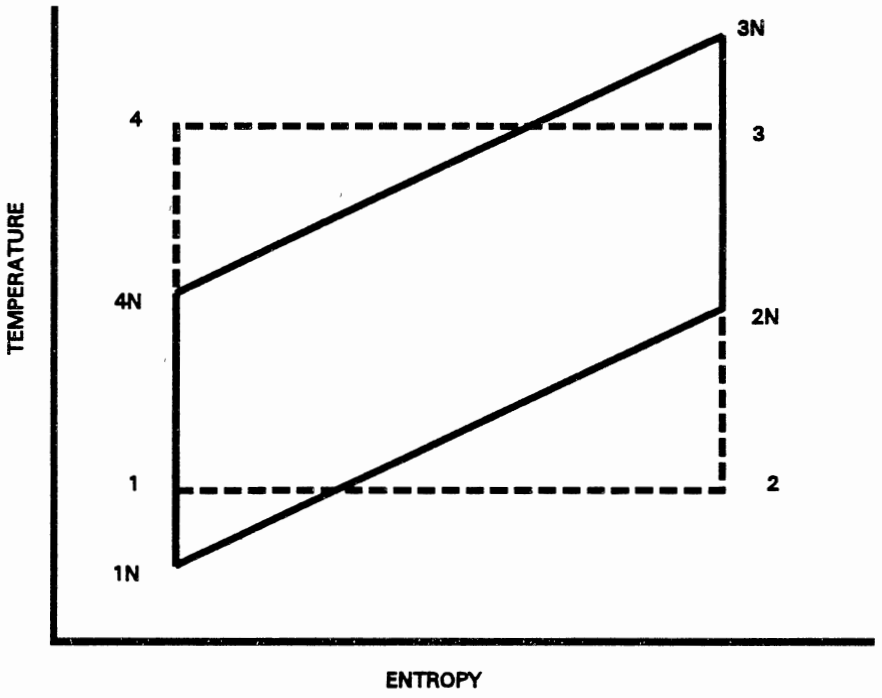


Figure 13. Temperature-Entropy Diagram of NARM and Pure Refrigerant Cycles

pure refrigerant is a Carnot cycle, while the cycle for the mixture is a Lorenz cycle. A measure of refrigerating efficiency is the coefficient of performance. It is defined as the area under the cycle divided by the area of the cycle , the coefficients of performance would be calculated as

$$COP_{Carnot} = \frac{\text{Area under 1-2}}{\text{Area (1-2-3-4)}} \qquad (2.45)$$

$$COP_{Lorenz} = \frac{\text{Area under 1N-2N}}{\text{Area (1N-2N-3N-4N)}} \qquad (2.46)$$

performance, the condensing temperature change of the NARM in the condenser allows a large temperature change in the heat sink fluid to be thermodynamically acceptable, and greatly reduced heat sink flows are allowed for a given heat rejection. This results in reduced the power consumption required to pump the heat sink flow through the condenser. In addition, it has been shown that mixtures of refrigerants can result in lower volumetric flowrates and pressure ratios for a given application, resulting in reduced compressor size. However, the heat transfer coefficient for NARM may be lower than for pure refrigerants.

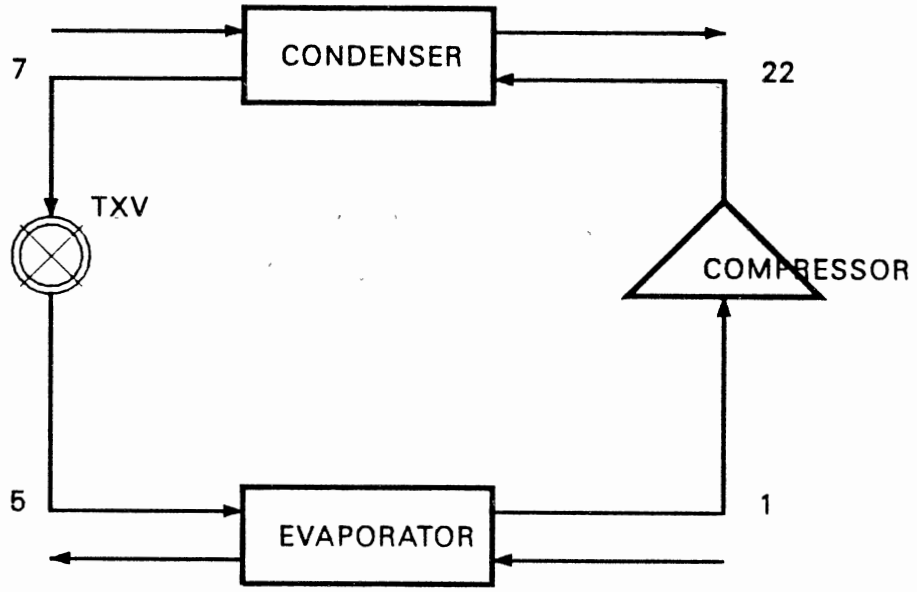
2.4 Description of NARM Vapor Compression Cycles

A literature search of material related to this topic indicates that most of the published papers evaluate the performance of NARM vapor compression cycles based on a conventional cycle suitable for single refrigerants. Very few published papers evaluate the performance of special cycles for NARM's.

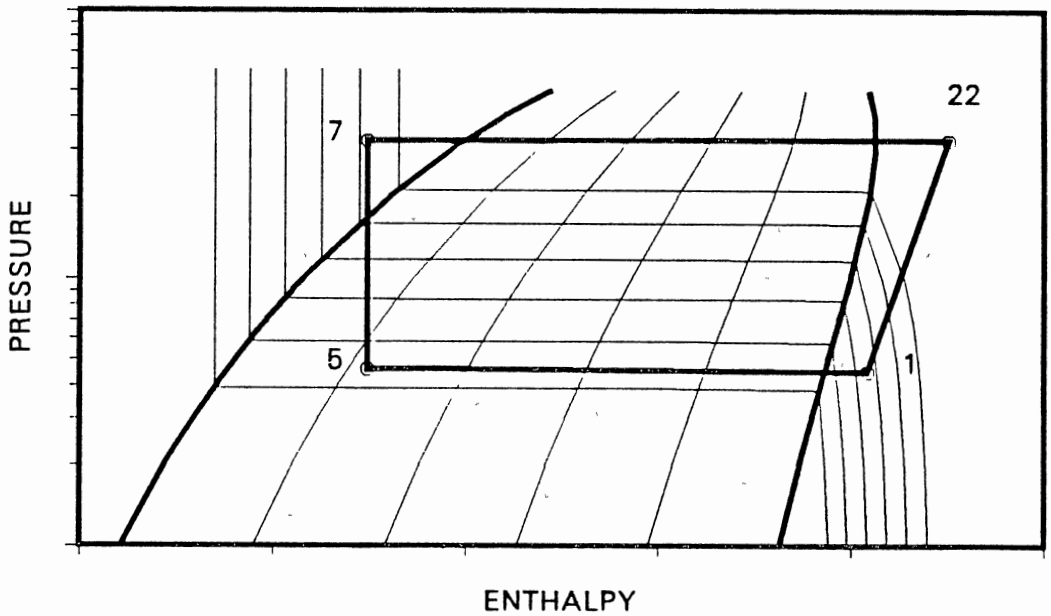
This study involves the arrangement of components in the thermodynamic cycle, comparative analysis of the performance and the hardware requirements for domestic and industrial applications for a wide range of acceptable refrigerant mixtures. To identify cycles and mixtures, six cycles are evaluated in this study. They are :

2.4.1 Cycle 1

The flow diagram is given in Figure 14a. This is a conventional cycle can operate with a pure refrigerant or an azeotropic mixture . The cycle is used as a base line to compare the performance of other



a. Cycle 1 Flow Diagram (Pure Refrigerant)



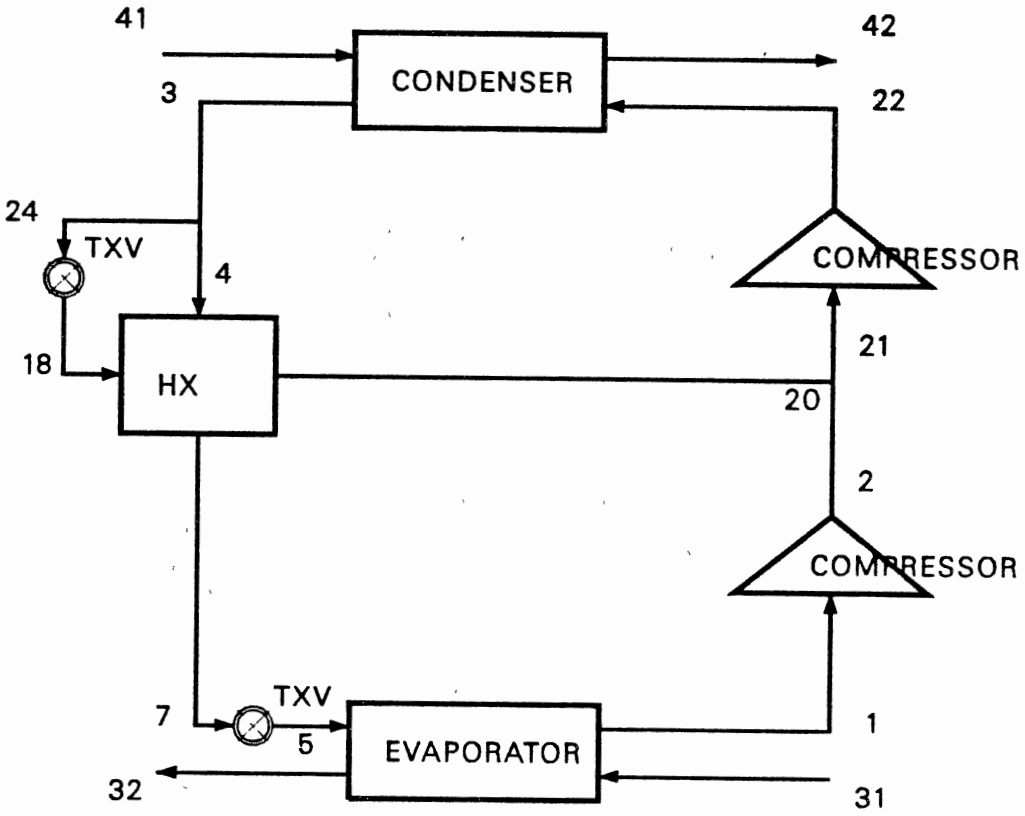
b. Cycle 1 P-H Diagram

Figure 14. Cycle 1 Flow Diagram and P-H diagram

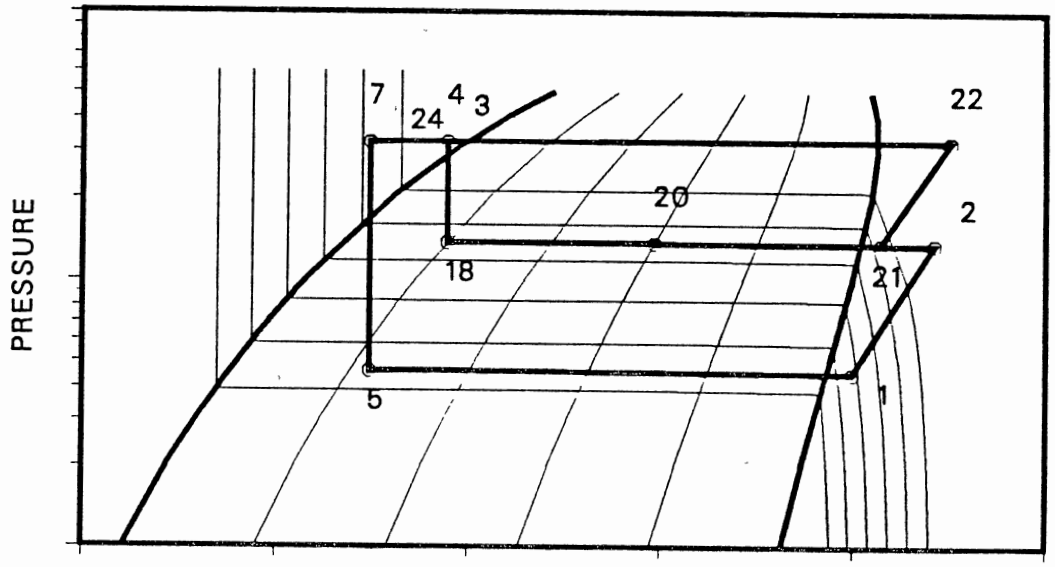
cycles. The conventional vapor compression refrigeration system has an evaporator where energy from the source heat load boils and slightly superheats the refrigerant at essentially constant pressure (state point 1). The superheated vapor is compressed to a temperature (and pressure) level (state point 22) from which heat from the evaporator and compression process can be rejected to a heat sink by the condenser at constant pressure, desuperheating, condensing, and slightly subcooling the working fluid (state point 7). From the subcooled condition, the refrigerant is flashed to evaporator pressure (state point 5) across an expansion valve to complete the cycle. The cycle state points on a pressure-enthalpy diagram (P-H diagram) are shown in Figure 14b for a pure refrigerant or azeotropic mixture refrigerant.

2.4.2 Cycle 2

The flow diagram is given in Figure 15a. This cycle uses a pure refrigerant or an azeotropic refrigerant mixture as the working fluid. An improvement over the base line cycle is achieved by including a direct expansion subcooler HX operating at interstage pressure. The subcooler significantly subcools the condensate leaving the condenser (state point 4) by using a part of the condensate (state point 24), flashed to interstage pressure (state point 18) as the heat sink. The subcooled refrigerant liquid leaving the subcooler HX (state point 7) flows through the expansion device undergoing a drop in pressure and temperature. The low pressure, low temperature, and low quality refrigerant enters the evaporator (state point 5), picks up heat from the low temperature heat source, reaching superheated vapor at the



a. Cycle 2 Flow Diagram (Pure Refrigerant)



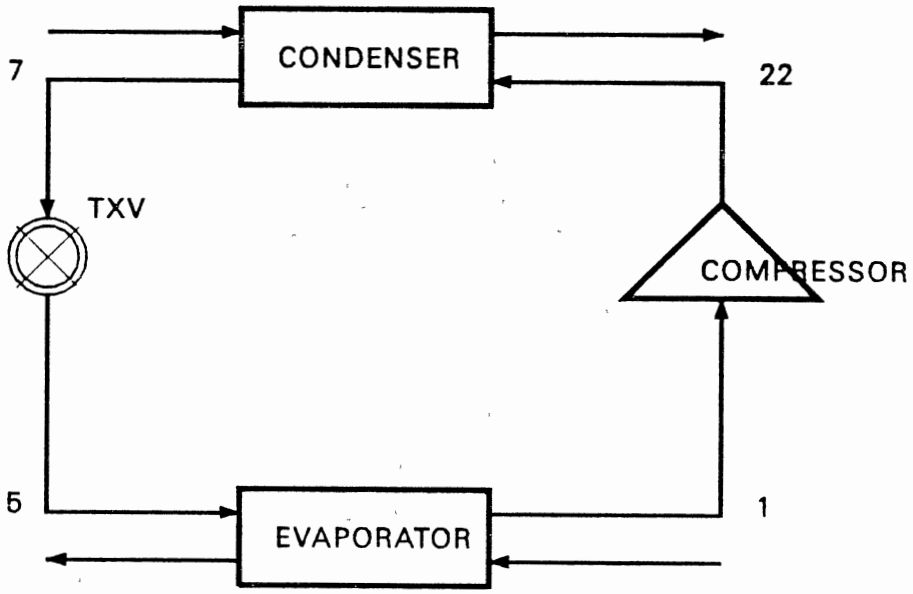
b. Cycle 2 P-H Diagram

Figure 15. Cycle 2 Flow Diagram and P-H Diagram

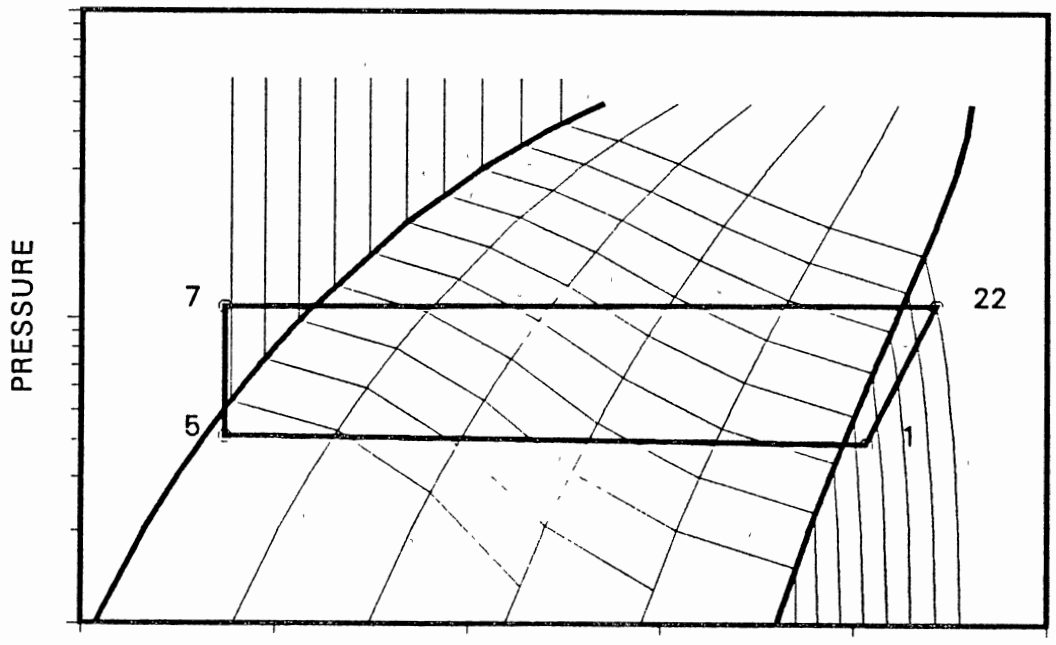
evaporator exit (state point 1). The superheated vapor at the evaporator exit is then compressed to interstage pressure (state point 2) by the first stage of the compressor and mixes with the low temperature two-phase refrigerant exit from the subcooler HX (state point 20) to give a slightly superheated vapor at the suction inlet of the second stage of the compressor (state point 21). The mixed superheated refrigerant is compressed by the second stage of the compressor to a temperature (and pressure) level (state point 22) from which heat from the evaporation and compression processes can be rejected to a heat sink by the condenser at constant pressure, desuperheating, condensing, and slightly subcooling the working fluid (state point 4). The cycle state points on the P-H diagram are shown in Figure 15b. Thermodynamically, the subcooler HX functions in a manner similar to that of a vapor liquid separator by removing the essentially saturated vapor existing at interstage conditions, which cannot contribute effectively to heat absorption in the evaporator, and directing it to the second compression stage. Thus, less flow at lower quality is admitted to the evaporator, and less work is required in the cycle. This arrangement is available only in multistage compressor designs.

2.4.3 Cycle 3

The arrangement of Cycle 3 is similar to Cycle 1 except a NARM is the working fluid for this cycle. The flow diagram is shown in Figure 16a and the state points on a P-H diagram are shown in Figure 16b.



a. Cycle 3 Flow Diagram (NARM)

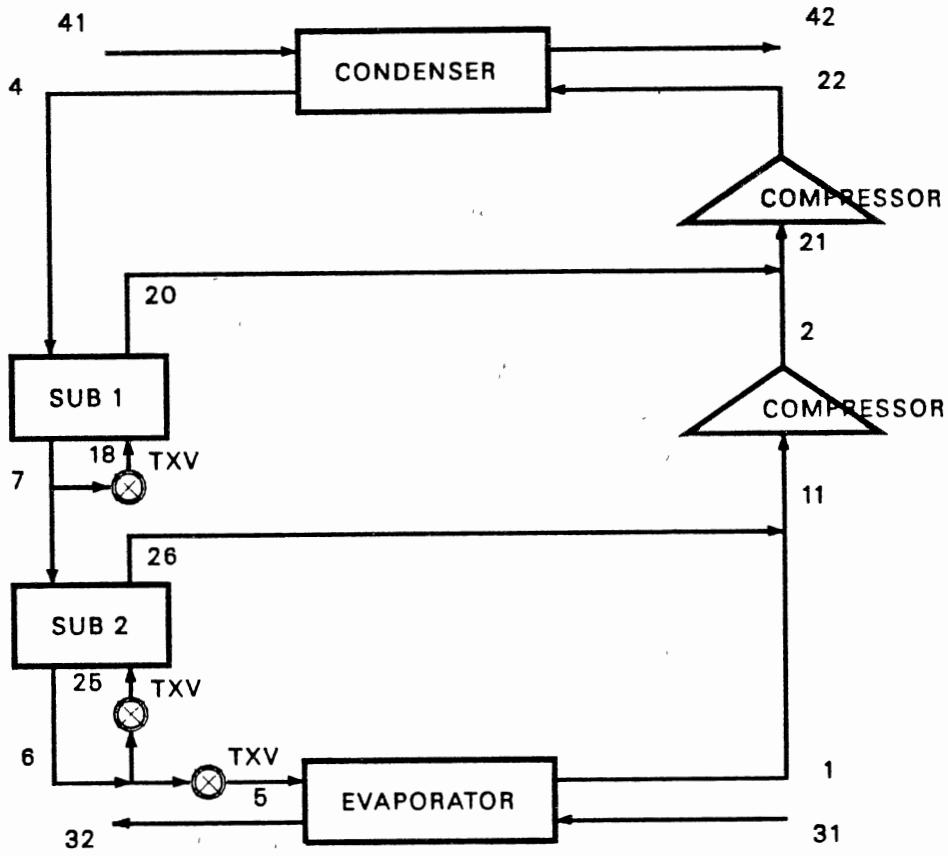


b. Cycle 3 P-H Diagram

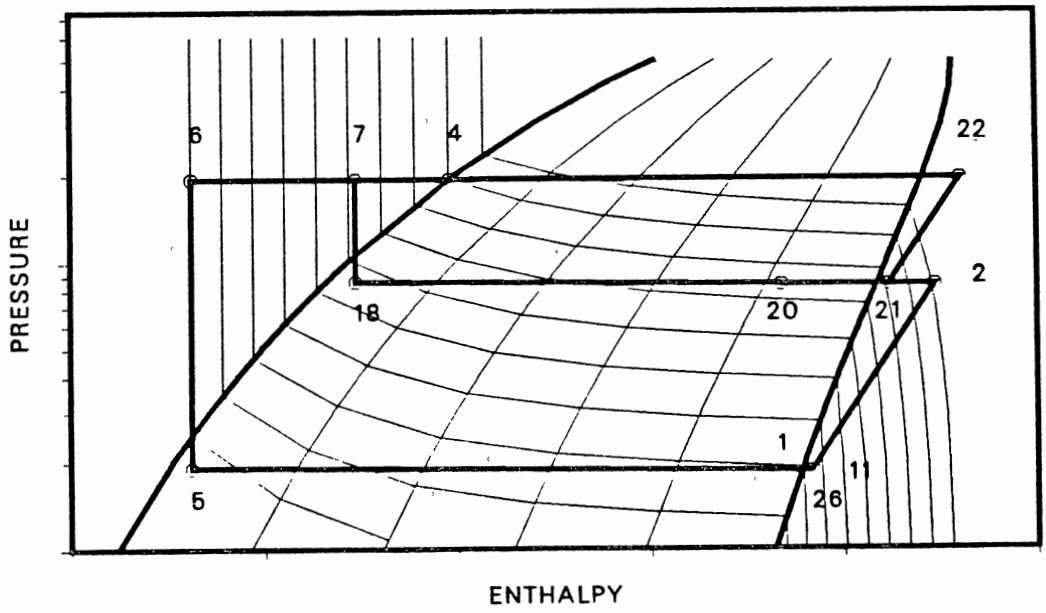
Figure 16. Cycle 3 Flow Diagram and P-H Diagram

2.4.4 Cycle 4

The thermodynamic configuration of the vapor cycle is shown in Figure 17a and was specifically designed for a system using a NARM. Liquid refrigerant leaving the condenser (state point 4) enters the subcooler HX I where it is further subcooled by evaporating two-phase refrigerant at approximately interstage pressure. Because the NARM evaporates with a rising temperature, the flow for this two-phase cooling is tapped from the subcooled liquid leaving the subcooler HX I (point 7), passes through an expansion device to a low temperature and interstage pressure (state point 18) and flows through the subcooler HX I which is of counterflow heat exchanger configuration . The two-phase NARM that leaves the subcooler HX I (state point 20) is then injected into the superheated vapor leaving the first compressor stage (state point 2). This interstage cooling increases the efficiency of the overall compression process and reduces the second stage outlet vapor temperature to within the safe operating limits for the refrigerants used in the NARM. The subcooled liquid leaving subcooler HX I (state point 7) enters subcooler HX II for further subcooling. The high pressure subcooled liquid leaving the subcooler HX II (state point 6) is divided into two streams, the main portion is flashed through an expansion device and routed to the evaporator (state point 5) and a small portion being used for two-phase cooling in subcooler HX II. The outlet of heat exchanger HX II (state point 26) is merged with the outlet of the evaporator (state point 1) to provide a minimum superheated vapor to the inlet of the compressor first stage (state point 11) and then compressed to interstage pressure (state point 2) by the first stage of the compressor. After mixing, the slightly



a. Cycle 4 Flow Diagram (NARM)



b. Cycle 4 P-H Diagram

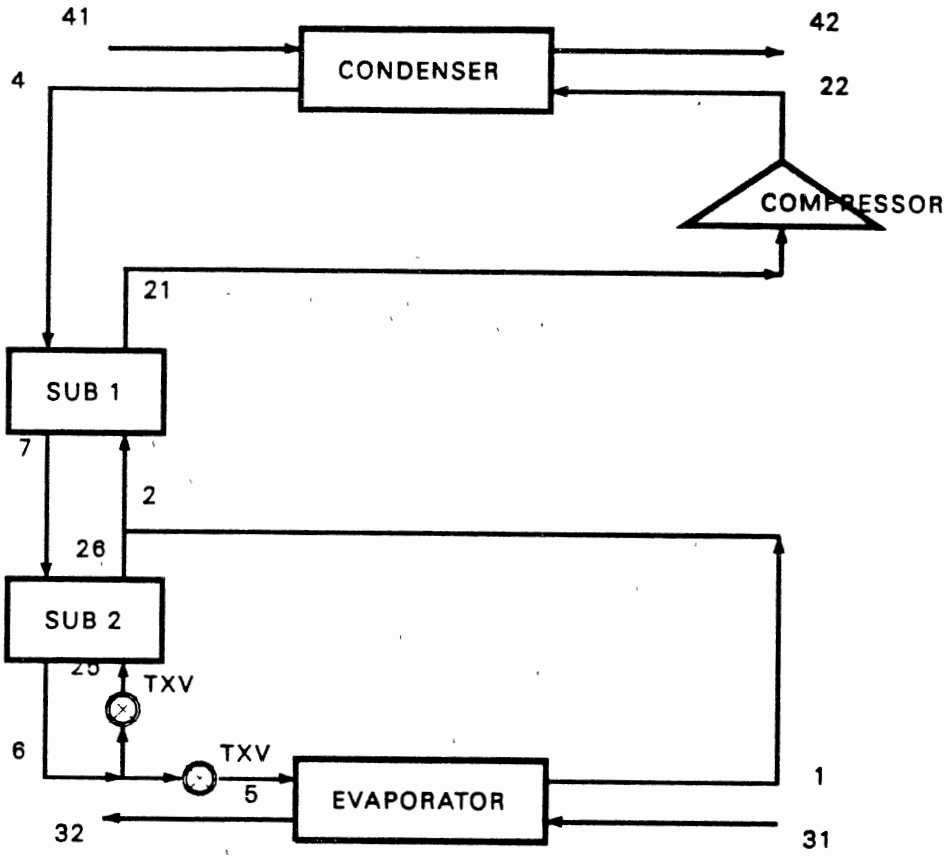
Figure 17. Cycle 4 Flow Diagram and P-H Diagram

superheated NARM (state point 21) is compressed by the second stage of the compressor to a temperature (and pressure) level (state point 22) from which heat from the evaporation and compression processes can be rejected to a heat sink by the condenser at constant pressure, desuperheating, and condensing the NARM to saturated liquid at the outlet of the condenser (state point 4) to complete the cycle. The cycle state points on P-H diagram are shown in Figure 17b.

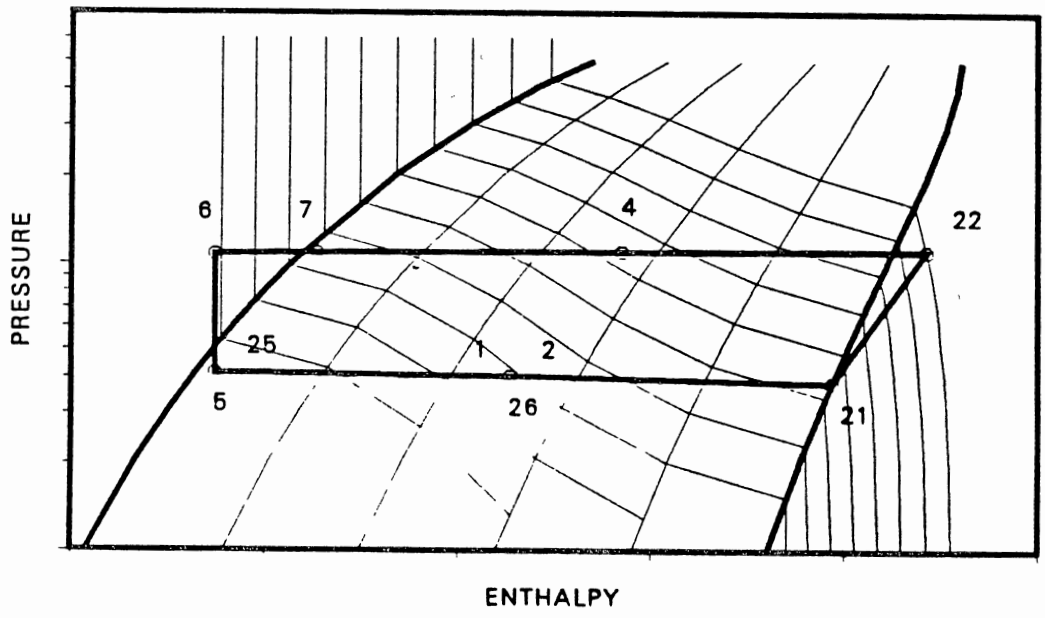
It should be noted that the subcooling obtained in subcooler HX II is of advantage only to a NARM system allowing a reduction in evaporator minimum temperature without a reduction in evaporator pressure. This is due to the slope of the constant temperature lines in the two-phase region of the NARM.

2.4.5 Cycle 5

This cycle is based on the fact that the isotherms in the two-phase region of certain NARM's are inclined with considerable downward slope to the isobars. The thermodynamic configuration of the vapor cycle is shown in Figure 18a and was specifically designed for a system using a NARM. Two-phase refrigerant leaving the condenser (state point 4) enters subcooler HX I where it is further cooled by evaporating two-phase refrigerant from the outlet of the evaporator (state point 2). The hot stream NARM leaving subcooler HX I at nearly saturated liquid (state point 7) enters subcooler HX II for further subcooling to the subcooled condition (state point 6). A portion of the subcooled refrigerant leaving subcooler HX II is flashed across an expansion device and returned to the other side of heat exchanger HX II and used for two-phase cooling in subcooler HX II. The remaining subcooled



a. Cycle 5 Flow Diagram (NARM)



b. Cycle 5 P-H Diagram

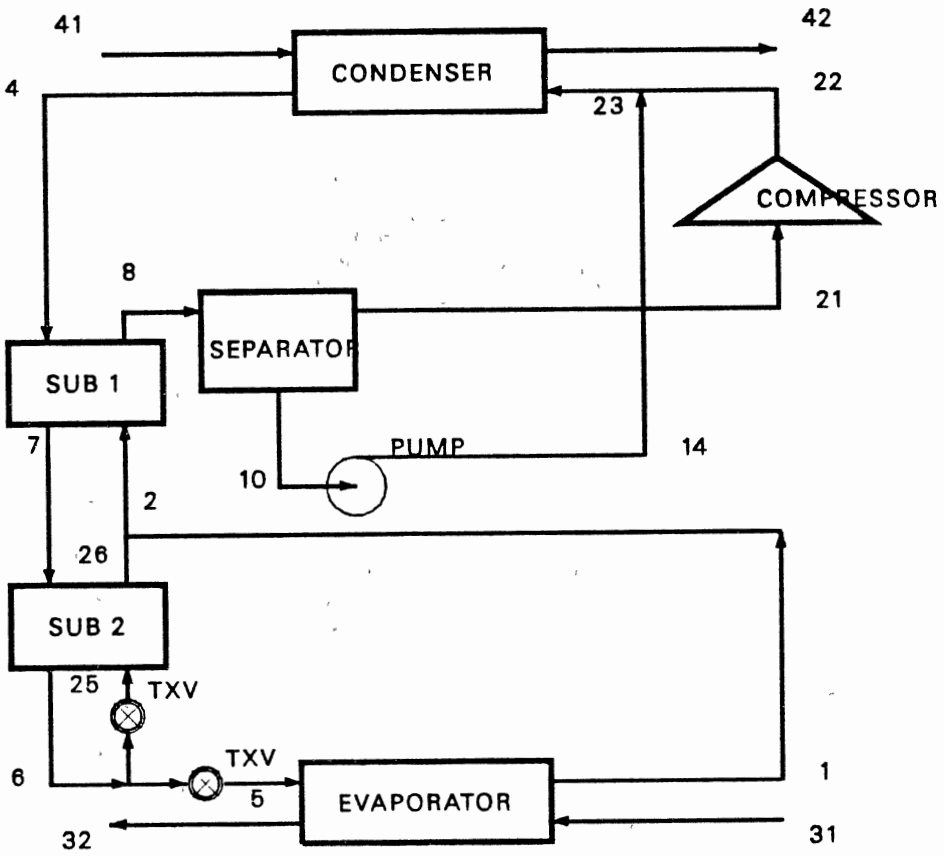
Figure 18. Cycle 5 Flow Diagram and P-H Diagram

refrigerant leaving subcooler HX II is then flashed to evaporator pressure across an expansion valve to the evaporator (state point 5). The selection of evaporator outlet quality and temperature is based on the heat source temperature. The refrigerant cold stream at the outlet of subcooler HX II (state point 26) joins with the refrigerant at the outlet of the evaporator (state point 1) and then the mixed flow (state point 2) passes though the subcooler HX I to get a minimum superheated vapor to the suction port of the compressor (state point 21). The superheated vapor is compressed to a temperature level (state point 22) at which heat can be rejected to a heat sink by the condenser to complete the cycle.The cycle state points on a P-H diagram are shown in Figure 18b

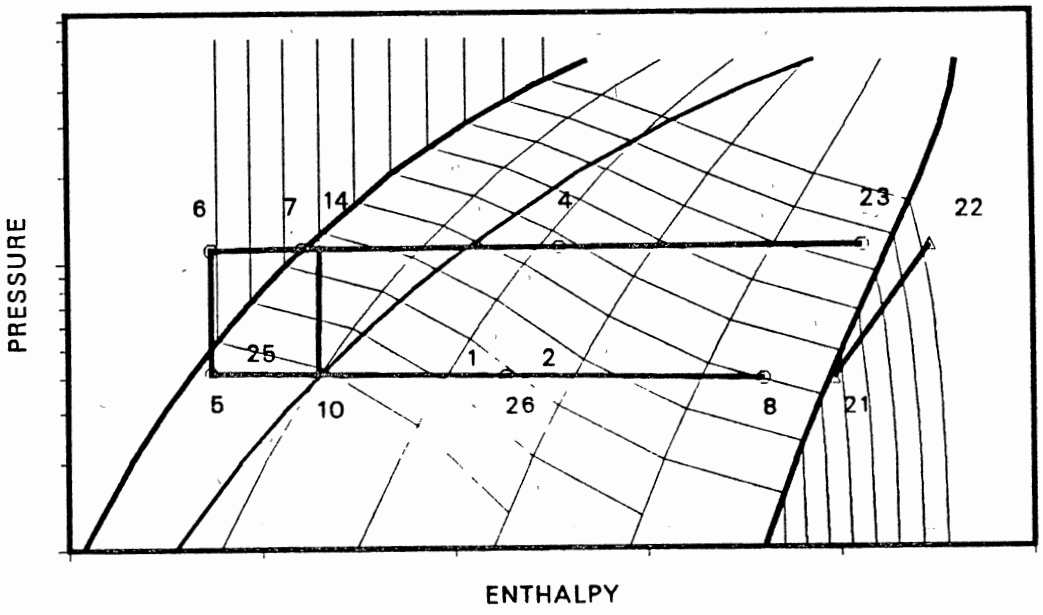
2.4.6 Cycle 6

The flow diagram is given in Figure 19a. Cycle 6 is similar to Cycle 5, except, instead of the cold NARM leaving the subcooler HX I (state point 8) as dry vapor it is now designed to leave as wet vapor. Wet vapor leaving the subcooler HX I passes though a separator-receiver, saturated vapor and liquid with different composition are separated, then the separated liquid (state point 10) is pumped to the pressure level of the condenser inlet (state point 14) and the separated vapor (state point 21) is compressed and then mixed with the separated liquid at the inlet of the condenser (state point 23). The cycle state points on a P-H diagram are shown in Figure 19b.

The characteristics of the cycles are summarized in the Table II.



a. Cycle 6 Flow Diagram (NARM)



b. Cycle 6 P-H Diagram

Figure 19. Cycle 6 Flow Diagram and P-H Diagram

TABLE II

SUMMARY OF CYCLES

	Cycle 1	Cycle 2	Cycle 3	Cycle 4	Cycle 5	Cycle 6
Working Fluid	Pure	Pure	NARM	NARM	NARM	NARM
N° Compressor Stage	1	2	1	2	1	1
N° Expansion Device	1	2	1	3	2	2
Evaporator Inlet Outlet	Dry 2-phase Sat/Sup vapor	Dry 2-phase Sat/Sup vapor	Dry 2-phase Sat/Sup vapor	Dry 2-phase Sat/Sup vapor	Flooded 2-phase 2-phase	Flooded 2-phase 2-phase
Condenser Inlet Outlet	Sup. Sat/Sub liquid	Sup. Sat/Sub liquid	Sup. Sat/Sub liquid	Sup. Sat/Sub liquid	Sup. 2-phase	Sup. 2-phase
Additional Heat Exchangers	0	1	0	2	2	2
Liquid Pump	0	0	0	0	0	1

Sat/Sup : Saturated or superheated vapor

Sat/Sub : Saturated or Subcooled liquid.

CHAPTER III

NARM SELECTION

3.1 Requirements of Refrigerants

A desirable refrigerant should possess the chemical, physical, and thermodynamic properties that provide the potential for high thermodynamic efficiency within the constraints of the heat source and heat sink temperature levels, within which the vapor compression cycle must operate. Several criteria are considered in the process to select pure refrigerants as well as refrigerant mixtures. These criteria can be broken down into three groups: safety and environmental requirements, physical and chemical characteristics, and thermodynamic characteristics.

3.1.1 Safety and Environmental Requirements

- Toxicity: There should be no danger to health or property due to leaks or other malfunctions in a refrigerating system.

- Flammability: For safety reasons, flammable fluids should be excluded from consideration, depending on the application and necessary safety precautions .

- Ozone Depletion Potential (ODP) and Global Warming Potential (GWP): This new environmental requirement must now be added to the traditional health and safety criteria. A refrigerant should not

contribute to ozone depletion, low level smog formations or greenhouse warming.

3.1.2 Physical and Chemical Characteristics

- The dielectric strength of the vapor should be high. This permits its use in hermetically sealed compressors where vapor comes in contact with motor windings.

- Instability: Breakdown of the refrigerant generally occurs at the compressor discharge port where the discharge temperature is the hottest part of the working cycle. The selection of a working fluid of a heat pump is more critical than the selection of a fluid for refrigeration or air conditioning, because heat pumps must be run at temperatures which are sometimes substantially higher. Therefore, chemical stability is of great importance to the heat pump designer.

- Heat transfer characteristics: The refrigerant should have thermophysical properties causing high heat transfer coefficients. These related properties are density, specific heat, thermal conductivity, and viscosity.

- Oil miscibility: The refrigerant should have good miscibility with oil, particularly at low temperatures.

3.1.3 Thermodynamic Characteristics

- The critical temperature: The critical temperature should be above the highest condensing temperature that might be encountered. With air-cooled condensers, in general, this would be above 130°F, and must always exceed the temperature of the heat sink.

- Latent heat of evaporation: A refrigerant should have a high latent heat of evaporation per unit of weight so that the amount of refrigerant circulated to produce a given refrigeration effect will be minimal. However, in small capacity systems, too low a flow rate may actually lead to problems such as the difficulty encountered in accurately controlling its flow through the regulating valve.

- Evaporating pressure: Evaporating pressures above atmospheric pressure is desirable to avoid leakage of moisture-laden air into the refrigerating system and permit easier detection of leaks. To satisfy this condition, the normal boiling temperature must be lower than the heat source temperature.

- Condensing pressure: Condensing pressure should be low to allow construction of light-weight equipment, which affects power consumption, compactness, and installation.

- Low freezing temperature: The refrigerant must not solidfy during normal operating conditions.

In addition to the above characteristics, the refrigerant should satisfy basic economic requirements, including commercial availability and low cost .

NARM's which are selected as the working fluids for VCS not only need to meet the above requirements for pure refrigerants but the refrigerant mixture should not form an azeotrope and require special thermodynamic properties to operate at high performance. In this study, the temperature range of the heat sink and heat source is limited to air conditioning, heat pump, and chiller applications.

In this section, emphasis is placed on the selection of binary mixtures based on the thermodynamic properties of mixtures.

3.2 NIST Refrigerants Database

In this study, working fluids of vapor compression cycles are chosen from the The National Institute of Standards and Technology (NIST) database. Currently the NIST Thermodynamic Properties of Refrigerants and Refrigerant Mixtures Database (version 2.0) (16) consists of 18 refrigerants and includes the promising refrigerants with zero or low depletion ozone potential such as R134, R134a, R142b, R123, and R124 to replace the refrigerants banned by the Montreal Protocol. Refrigerant mixtures selected as working fluids for vapor compression cycles need to show they can operate at higher performance levels compared to pure refrigerant systems. In order to evaluate the performance (energy efficiency, capacity etc...) of the cycles and working fluids, a complete set of thermodynamic properties is required.

The equation-of-state is used to estimate the enthalpy, entropy, density, and specific volume of liquid and vapor under saturated, subcooled, and superheated conditions. The Carnahan-Starling-DeSantis (CSD), Benedict-Webb-Rubin (BWR), Peng-Robinson, Lee-Kesler-Plocker (LKP), Redlich-Kwong-Soave (RKS), and other models are used for this purpose. Each model has its advantages and disadvantages. The NARM properties developed by The National Institute of Standards and Technology (NIST) based on the Carnahan-Starling-DeSantis (CSD) equation of state.

The Carnahan-Starling-DeSantis (CSD) equation of state has been applied to refrigerants, particularly refrigerant mixtures. The attributes of a strong theoretical basis and relative simplicity allow the CSD equation to represent a fluid with minimum data. Morrison and McLinden have shown that saturation data alone (i.e., liquid and vapor

density and vapor pressure) produce nearly the optimal function for correlating all the thermodynamic properties. They have also shown that the CSD equation of state does not represent the region near the critical point well (37). However, this deficiency does not affect the application to vapor compression cycles because the cycles do not operate near the critical region.

Eighteen refrigerants in the NIST database are given in Table III and Figure 20. A comparison of Ozone-Depletion Potential (ODP) and Global Warming Potential (GWP) of the refrigerants is also shown in Table III. The data of this table found from the NASA Panel for Scientific Assessment.

The pure refrigerant properties found from (16),(27),(29),and (52). Some characteristics of main refrigerants are:

R-11 has a boiling point of 74.9°F and has wide usage as a refrigerant in indirect industrial and commercial air conditioning systems employing single or multistage centrifugal compressors. The refrigerant is also used in centrifugal chillers for commercial buildings. R-11 is banned because of the concern about the depletion of the ozone layer.

R-12 has a boiling point of -21.7°F, and has been used as a refrigerant in domestic refrigerators for more than 60 years. The refrigerant has suitable thermodynamic properties, chemical stability, non-toxicity, non flammability, and low cost. R-12 is also used in freezers, automobile air conditioners, refrigerated vending machines, food display cases, and a variety of small home and business appliances (e.g., dehumidifiers, water fountains). It is also used in centrifugal

TABLE III

REFRIGERANTS IN THE NIST DATABASE

	REFRIGERANT NAME	CHEMICAL FORMULAR	CRITICAL		ODP	GWP
			TEMP (F)	PRESS (Psia)		
R11	Trichlorofluoromethane	CCl3F	388.4	639.5	1.	1.
R12	Dichlorodifluoromethane	CCl2F2	233.6	596.9	.9-1.0	2.8-3.4
R13	Chlorotrifluoromethane	CClF3	83.9	561.0		
R13B1	Bromotrifluoromethane	CBrF3	152.6	575.0	10.0	
R14	Tetrafluoromethane	CF4	-50.2	543.0	0.0	
R22	Chlorodifluoromethane	CHClF2	204.8	721.9	.04-.06	.32-0.37
R23	Trifluoromethane	CHF3	78.1	701.4	0.0	
R113	1,1,2-trichlorotrifluoroethane	CCl2FCClF2	417.4	489.9	0.83	1.46
R114	1,2-dichoro-2,2,2-trifluoroethan	CClF2CClF2	294.3	473.0	0.71	4.92
R115	Chloropentafluoroethane	CClF2CF3	175.9	457.6	0.3-0.5	7.4-7.6
R123	1,1-Dichloro-2,2,2-trifluoroethane	CHCl2CF3	363.0	523.0	.013-.022	.017-.02
R124	1-chloro-1,2,2,2-tetrafluoroethane	CHClFCF3	294.3	514.0	0.02	0.10
R125	Pentafluoroethane	CHF2CF3	154.0	499.0	0.0	
R134	1,1,2,2-tetrafluoroethane	CHF2CHF2	237.0	497.0	0.0	0.0
R134a	1,1,1,2-tetrafluoroethane	CH2FCF3	214.0	589.8	0.0	.24-0.29
R142b	1-chloro-1,1-difluoroethane	CH3CClF2	278.8	598.0		
R152a	1,1-difluoroethane	CHF2CH3	236.3	652.0	0.0	0.02
RC270	Cyclopropane	CH2CH2CH2	256.7		0.0	

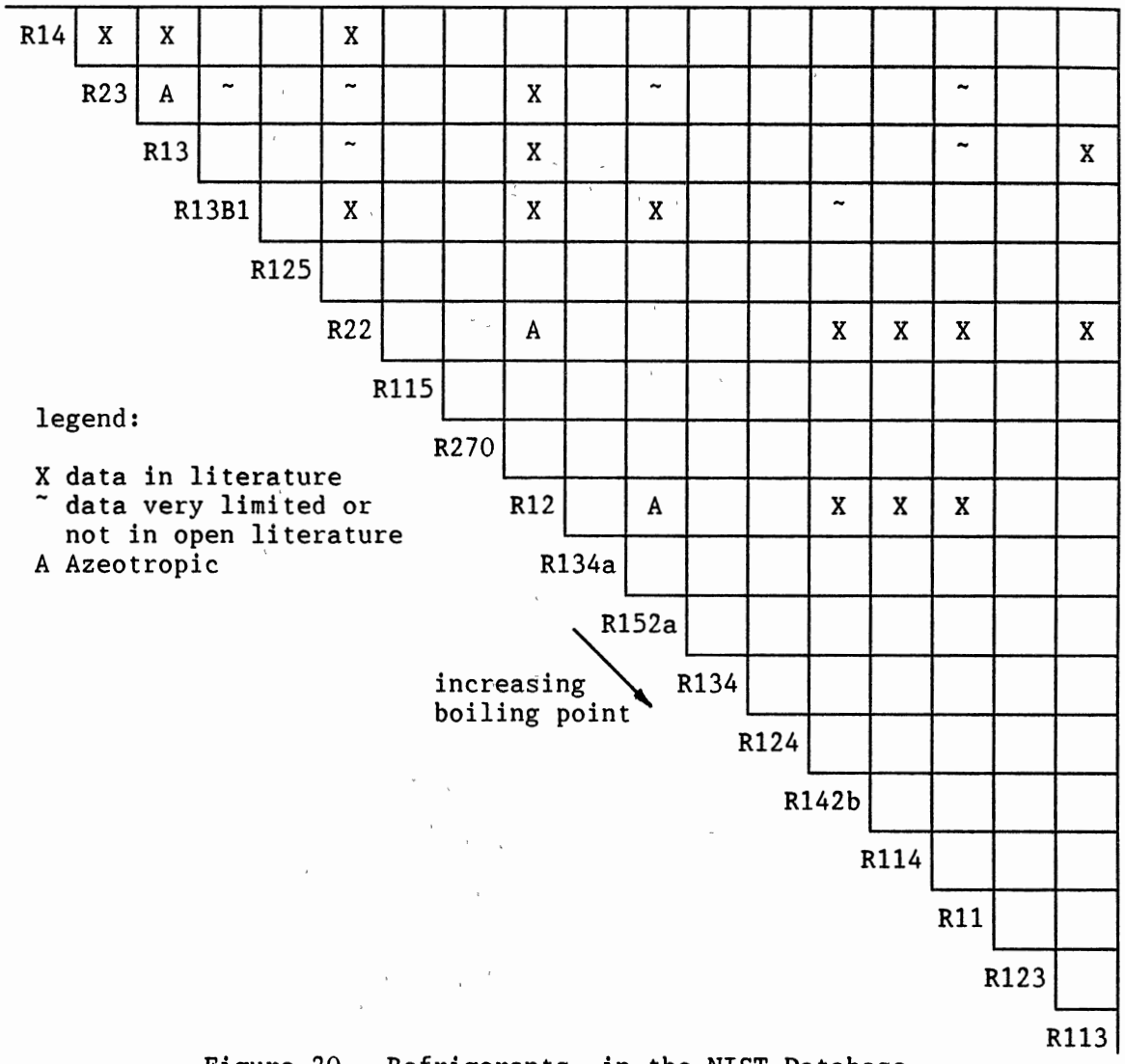


Figure 20. Refrigerants in the NIST Database

chillers for commercial buildings. R-12 has also been banned because of the concern about the depletion of the ozone layer.

R-13 has a boiling point of -114.6°F and is used in low-temperature specialty applications employing reciprocating compressors and generally in cascade with R-12 or R-22.

R-14 has boiling point of -198.3°F. Because R-14 has a very low critical temperature (-50.2°F), pure R-14 cannot be selected as a working fluid in this study.

R-22 has a boiling point of -41.4°F. This refrigerant is widely used in chillers, air conditioners, and heat pumps for domestic and commercial applications. R-22 has not been used for refrigerators and freezers. The outstanding thermodynamic properties of R-22 permit the use of smaller equipment than the sizes required by similar refrigerants, making it especially suitable when minimal size is essential. R-22 is not regulated by the Montreal Protocol, as it has not yet been considered a significant part of the ozone depletion problem by the Protocol or the EPA. However, several bills pending in Congress have been introduced to regulate R-22.

R-113 has a boiling point of 117.6°F. It is used in commercial and industrial air-conditioning as well as process water and brine cooling with centrifugal compression. The refrigerant is especially useful in small-tonnage applications. R-113 is also banned by the Montreal Protocol.

R-114 has a boiling point of 38.5°F. For high condensation temperatures, R-114 is considered an extremely stable refrigerant. R-114 is used as a refrigerant in fractional-horsepower household refrigerating systems and drinking-water coolers employing rotary vane-type compressors. The refrigerant is also used in industrial and commercial refrigerating systems to cool industrial process water and brine to -70°F, employing cascaded refrigeration systems with multistage centrifugal-type compressors. These systems involve 100 tons

refrigerating capacity and larger. It is also used in shipboard chillers for the Navy. R-114 is also banned.

R-115 has a boiling point of -37.7°F . The refrigerant is especially stable, offering a particularly low discharge temperature in reciprocating compressors. Its capacity exceeds that of R-12 by as much as 50 percent in low temperature systems. R-115 is also banned.

R-123 has a boiling point of 81.7, and has been suggested as a substitute for R-11. However, R-123 is more toxic than R-11. R-123 is non flammable.

R-124 has a boiling point of 10.4°F and is the leading candidate to replace R-114 as a refrigerant for marine refrigeration and stationary chillers. With a low ODP and GWP, R-124 offers attractive environmental properties.

R-125 has a boiling point of -55.3°F. R-125 may possibly be used for low temperature applications. Little information is available for R-125.

R-134 has a boiling point of -3.5°F and is non-flammable. It is currently under development. Studies indicate R-134 has better thermodynamic properties (for a conventional vapor compression cycle) than R-134a.

R-134a has a boiling point of -15.7°F. As a new refrigerant, R-134a has been suggested as a substitute for R-12 in refrigerators and chillers. The characteristics of R-134a include the zero value of ODP, non-flammability and stability. The temperature of the boiling point and critical point are very close to that of R-12. However, development risks include compatibility issues with oils and materials. In

addition, the toxicity of this substitute has not been fully investigated.

R-142b has a boiling point of 14.4°F, and is one of the new refrigerants recently made available on the market . R-142b may be used for medium temperature applications. The refrigerant is also suggested to replace R-12 and has the same advantages as R-134a. However, R-142b is moderately flammable. The application of a flammable refrigerant with good thermodynamic properties is still being discussed in the literature and by manufacturers.

R-152a has a boiling point of -13.°F. R-500 was invented in 1950, and is comprised of R-12 and R-152a (26.2 percent R-152a by weight) to create an azeotropic refrigerant mixture. Currently, R-152a is suggested to replace R-12 and also has the same advantage as R-134a. However, R-152a is similar to R-142b in that it is moderately flammable.

RC270 has a boiling point of -28.3°F. Flammable cyclo-propane is also a candidate to replace R-12 in refrigeration and automotive applications.

3.3 NARM Selection Based on Thermodynamic Properties

As mentioned earlier, to reduce the irreversibility to get the most benefit out of NARM vapor compression cycles, the temperature difference between the source side and NARM side in the evaporator must be designed constant or nearly constant. That is, the temperature profiles of the source and the NARM through the evaporator should be parallel, with the temperature difference is as small as possible. To provide a fair comparison between the cycle and the working fluid, the pinch point temperature of the evaporator is assumed to be at least

5°F. In this study, applications are considered with the source outlet temperature kept at 40°F and the source temperature loss is assumed 15°F and 30°F. From the above conditions, the maximum inlet temperature of NARM will be 35°F.

Like the evaporator, the temperature difference between the sink side and NARM side in the condenser must be kept nearly constant and as small as possible. In this study, applications are considered with the sink temperature at 100°F. The sink flow rate can be increased or reduced to get nearly parallel temperature profile through the condenser. The refrigerant condensing temperature can be selected to get at least 5°F pinch point temperature in the condenser. The power consumption to pump the sink fluid through the condenser is proportional with the sink flow rate for the same sink fluid pressure drop. Therefore the design sink flow rate effects to the overall system power consumption (total power consumption of the compressor, fan or pump source side and fan or pump for sink side)condenser system or of the pump for a liquid cooled condenser system.

There are two types of NARM evaporators: Type I and Type II.

- Type I (dry evaporator) is used for Cycle 3, and Cycle 4 which requires a saturated vapor or slightly superheated vapor at the outlet of the evaporator.

- Type II (flooded evaporator) is used for Cycle 5, and Cycle 6, which requires two-phase NARM exit at the outlet of the evaporator.

Each type of evaporator requires a different characteristic of NARM to operate with a high performance.

3.3.1 Type I Evaporator Thermodynamic Properties

For the Type I evaporator, the NARM changes from low quality two-phase at the inlet of the evaporator to saturated vapor or slightly superheated vapor at the exit of the evaporator. To reduce the irreversibility in the evaporator, a NARM must be selected with constituents and compositions such that its "gliding temperature interval" is equal to the temperature loss of the source. The "gliding temperature interval" is defined as the temperature difference between saturated vapor and saturated liquid at the working pressure. Figure 21 shows the gliding temperature interval of an A-B mixture as a function of composition of A. As shown in this figure, the gliding temperature intervals of these mixtures are less than the temperature loss of the source side and are not matched with the source side. Therefore, if these mixtures are used as the working fluid for the cycles with the Type I evaporator, the highest performance will be expected from the composition which yields the highest gliding temperature interval (composition is nearly 0.50A/0.50B). However, the cycles which operate with A-B mixtures, still cannot achieve the highest performance. Figure 22 shows the gliding temperature interval of a C-D mixture as a function of composition of constituent C. As shown in this figure, there are two NARM's with compositions C1 and C2 that have gliding temperature intervals equal to the source temperature loss. The gliding temperature interval of mixtures with the composition between zero to C1 and between C2 to 1.0 are less than the temperature loss of the source side. The behavior of these mixtures are similar to the mixtures A-B above. The pinch point temperatures occur at the inlet of the evaporator. The gliding temperature interval of mixtures with the

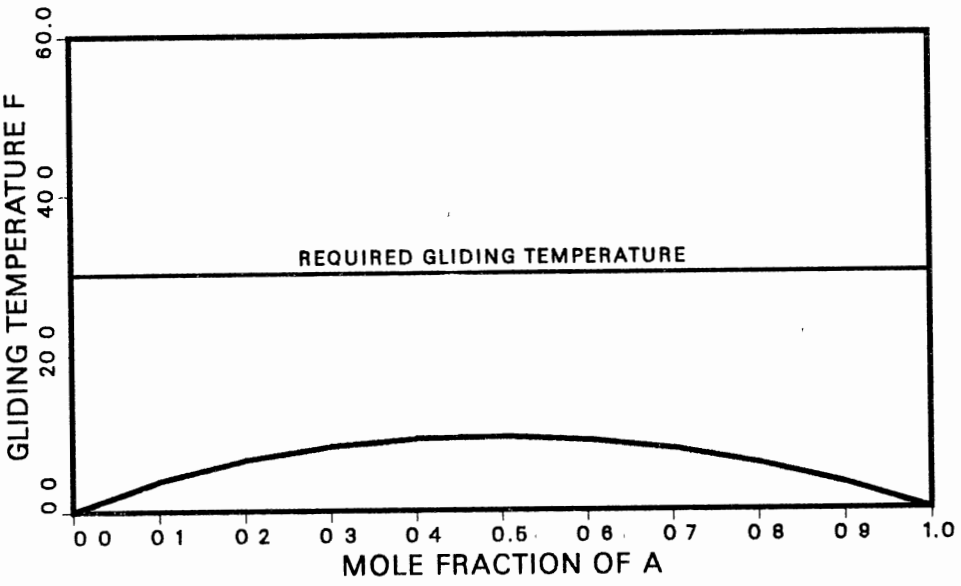


Figure 21. Gliding Temperature Interval of A-B Mixture

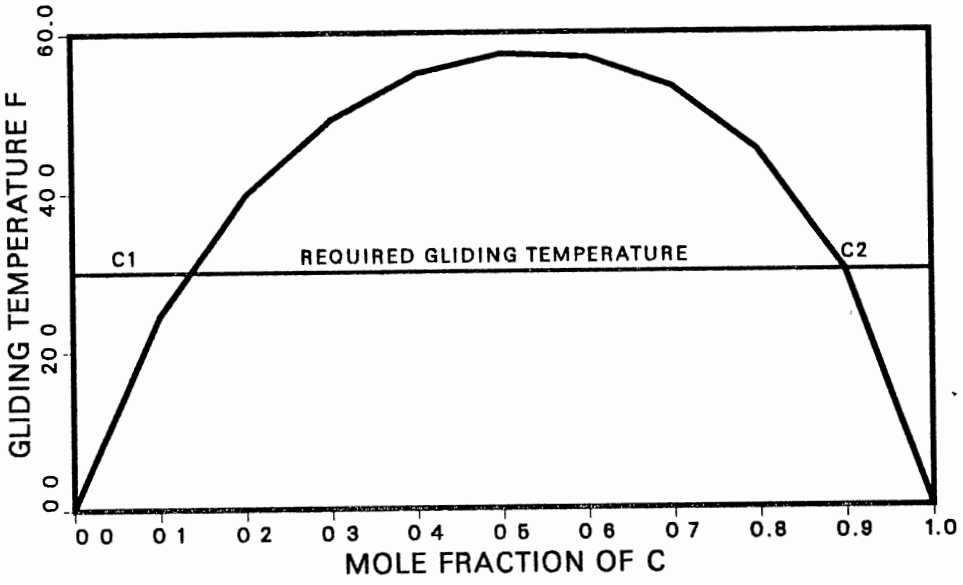


Figure 22. Gliding Temperature Interval of C-D Mixture

composition between C1 and C2 are greater than the temperature loss of the source side. If these mixtures are use for the Type I evaporator, the pinch point occurs at the exit of the evaporator and the average temperature difference between the source and the mixtures is greater than 5°F. The cycles will not operate at the highest possible performance. Using mixtures with compositions around C1 or C2 for the cycles with the Type I evaporator will have a nearly parallel temperature profile through the evaporator and satisfy the required pinch point temperature. These mixtures are expected to have the highest performance. The saturated pressure of liquid mixtures (preferred higher than atmospheric pressure) and latent heat of evaporation (preferred high) are also factors for selecting mixtures. To help select a possible NARM, saturated pressure of NARM liquid at 35°F, the latent heat of evaporation and the maximum gliding temperature interval of available NARM's from the NIST database are given in Figure 23 (for mixtures of R-11) and appendix A. Note that, on these figures, the mixtures with zero maximum gliding temperature are azeotropic mixtures. To eliminate NARM's which do not satisfy the gliding temperature requirement for the cycles with a Type I evaporator, a computer program has been written to select the possible non-azeotropic binary refrigerant mixtures (composition and constituents) for a specified temperature loss of the source side. The computer program flow chart is shown in appendix B. The result of the survey is given in Figure 24 for a 15°F gliding temperature, in Figure 25 for a 30°F gliding temperature, and in Figure 26 for a 60°F gliding temperature. The data in Figures 24, 25 and 26 are shown the saturated

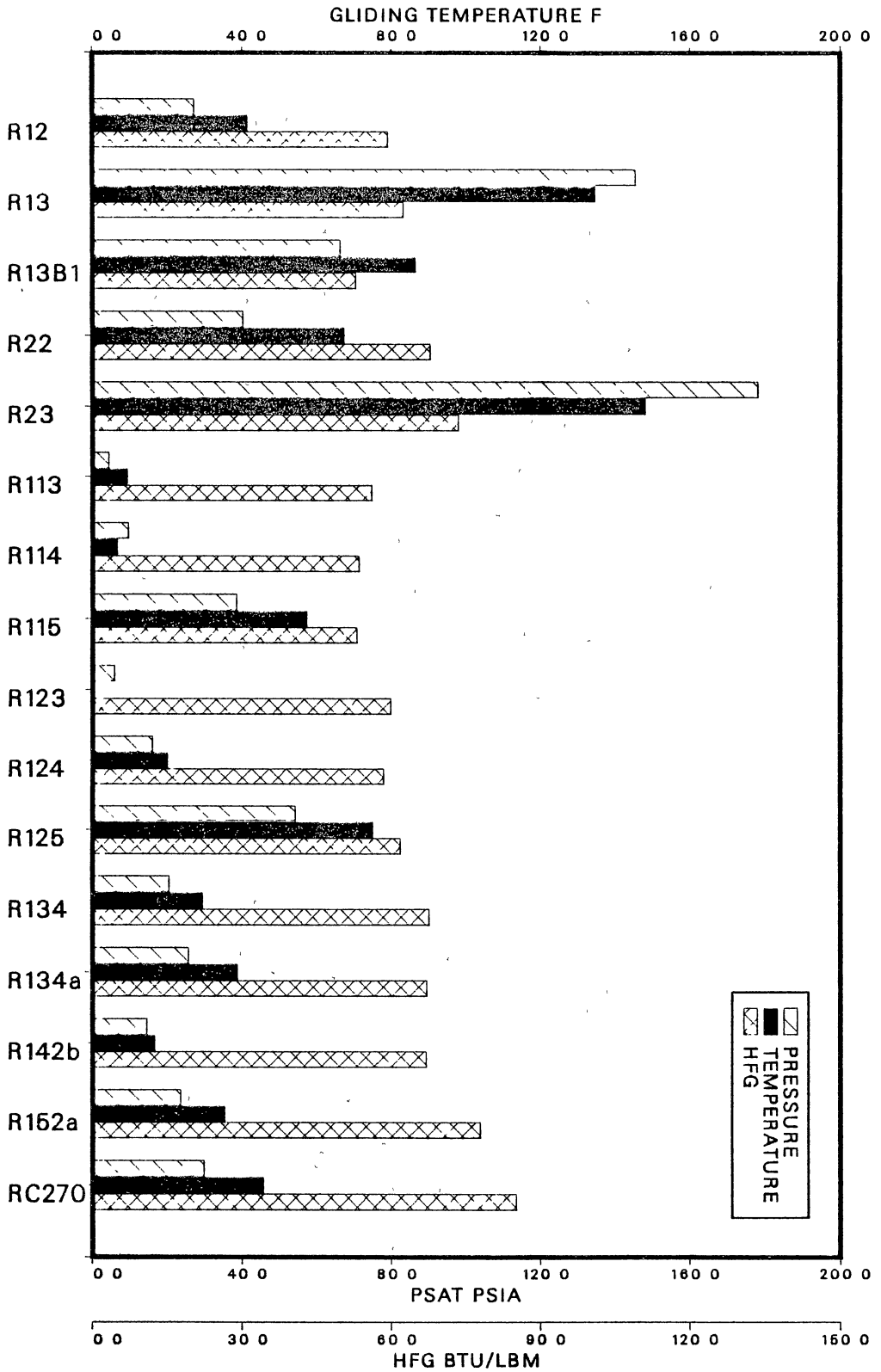


Figure 23. Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R11 Mixtures at 35F Sat. Liquid Pressure

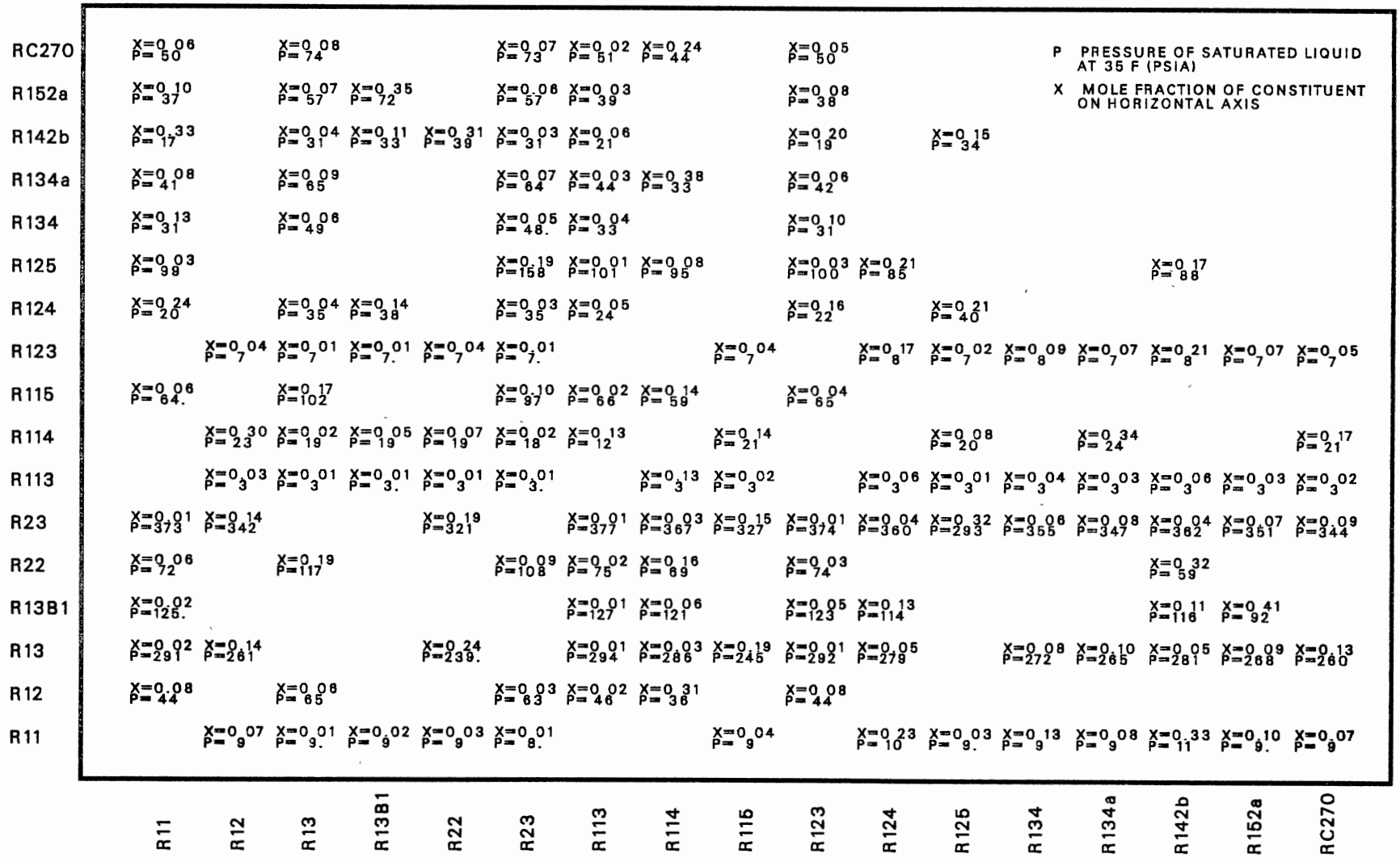


Figure 24. 35F Sat. Liquid Pressure, Mole Fraction of Binary Mixtures with 15F Gliding Temperature

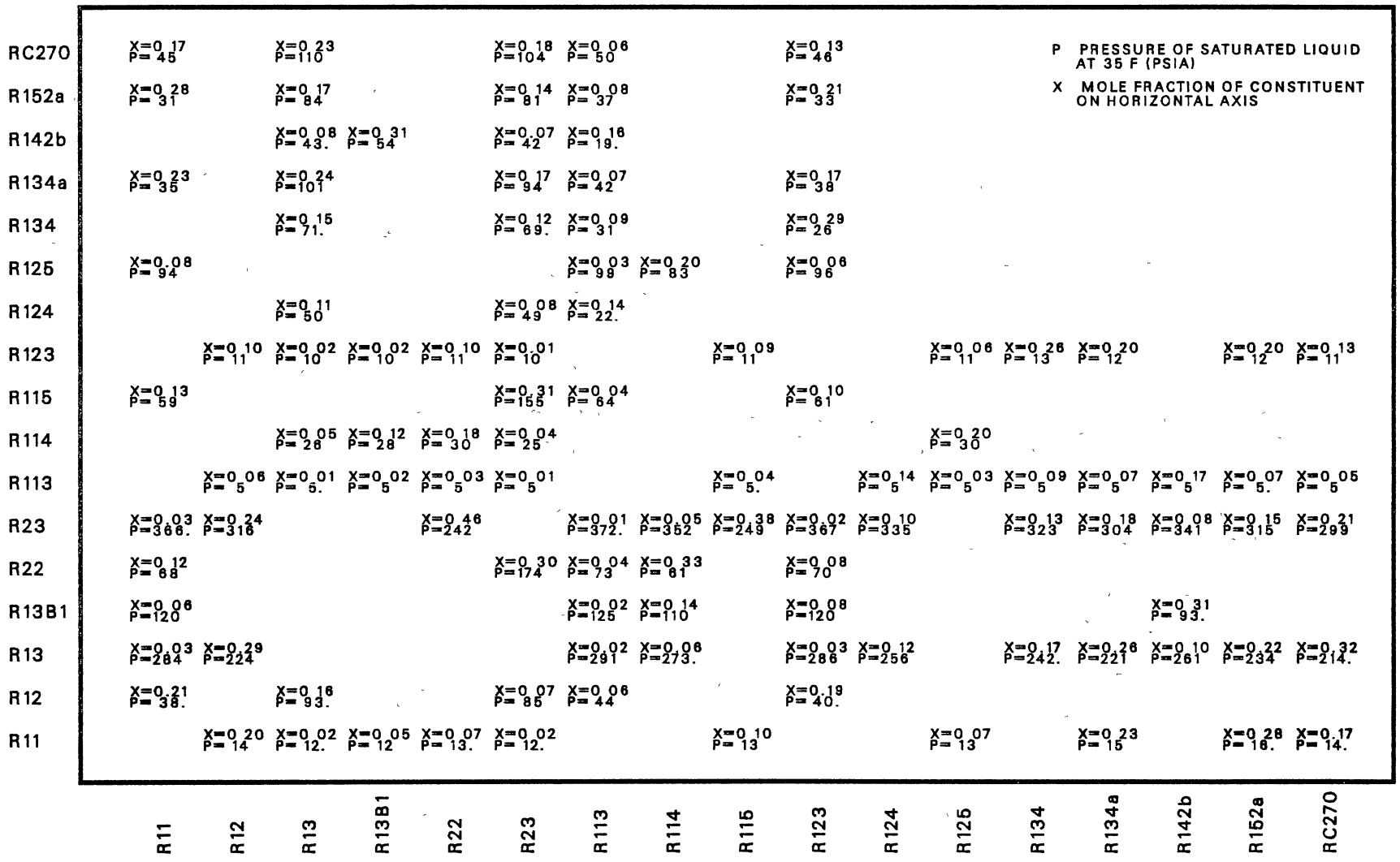


Figure 25. 35F Sat. Liquid Pressure, Mole Fraction of Binary Mixtures with 30F Gliding Temperature

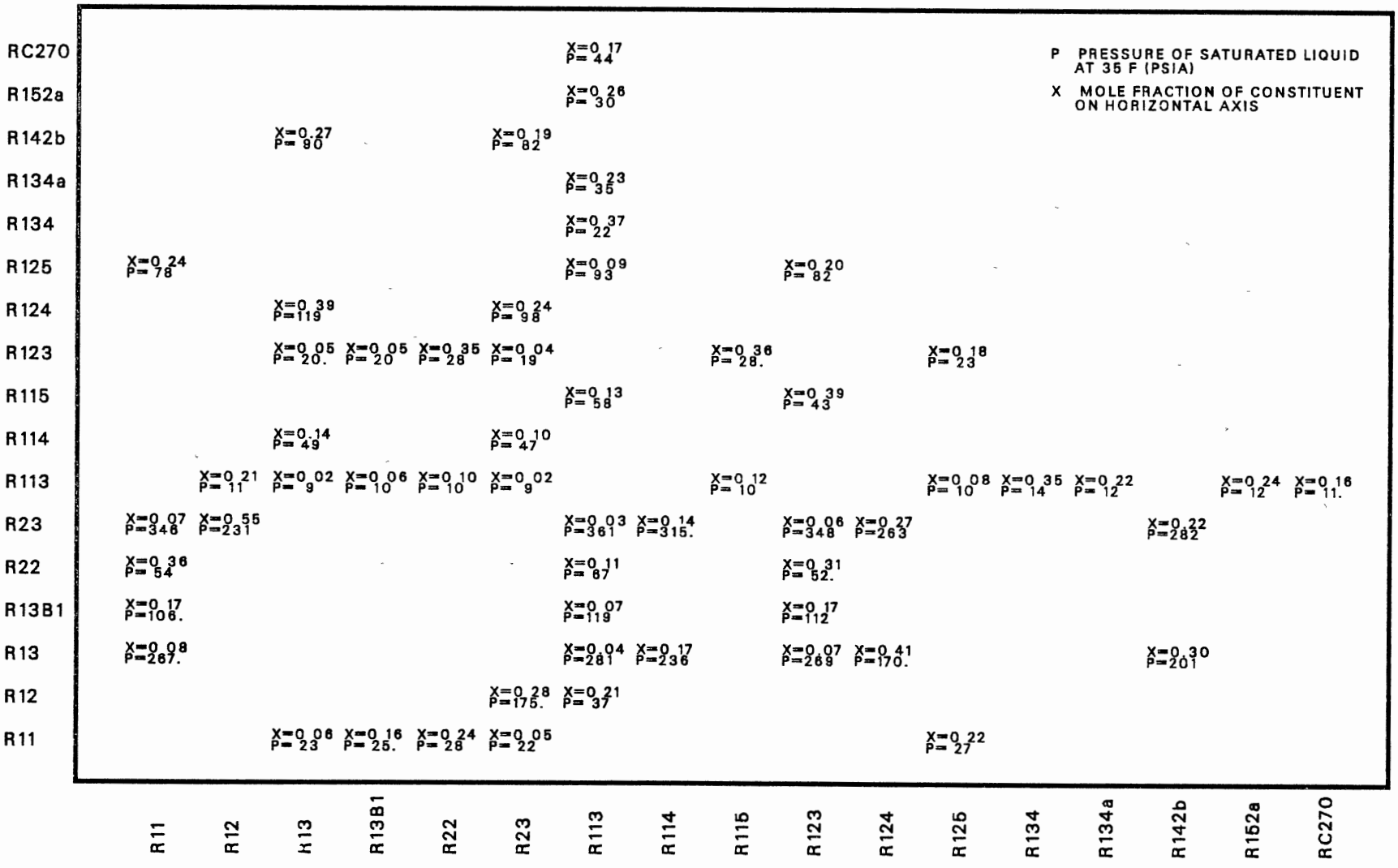


Figure 26. 35F Sat. Liquid Pressure, Mole Fraction of Binary Mixtures with 60F Gliding Temperature.

pressure of liquid NARM at 35°F and the mole fraction x of the constituent which indicated on the horizontal axis of these figures.

3.3.2 Type II Evaporator Thermodynamic Properties

For the Type II evaporator, the refrigerant mixtures change from low quality two-phase at the inlet of the evaporator to higher quality two-phase at the outlet of the evaporator (approximately 0.5). In this study, the term "0.5 quality gliding temperature" is defined as the difference of the NARM temperature between a saturated NARM liquid and a 0.5 quality NARM at the same pressure. For this type of evaporator, it is necessary to select a NARM (constituents, and compositions) and a design evaporator outlet quality to allow the gliding temperature of the NARM in the evaporator to be equal to the temperature loss of the source. Note that the two-phase NARM temperature change is non-linear with the quality. Figure 27 shows the temperature of an E-F non-azeotropic binary refrigerant mixture at working pressure as a function of quality for the mixtures with composition E1, E2, and E3 (E3 NARM has the highest gliding temperature interval). As shown in this figure, with exit quality 0.5, the 0.5 gliding temperatures of all mixtures are less than the temperature loss of the source side. Therefore, if these mixtures are used as the working fluid for the cycles with a Type II evaporator, the cycles will not operate at the highest possible performance. However, a NARM with a composition near E2 can be selected to operate for the cycles with a Type I evaporator. Figure 28 shows the temperature of a G-H non-azeotropic binary refrigerant mixture at working pressure as a function of quality for mixtures with composition G1, G2, G3, G4. As shown in this figure, at exit quality 0.5, the

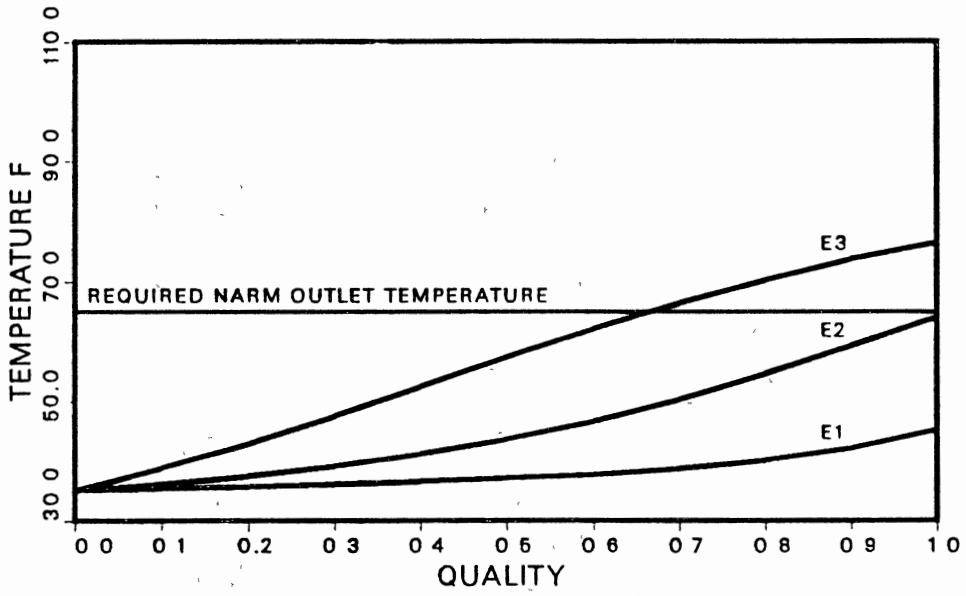


Figure 27. E-F NARM Temperature v/s Quality at Working Pressure

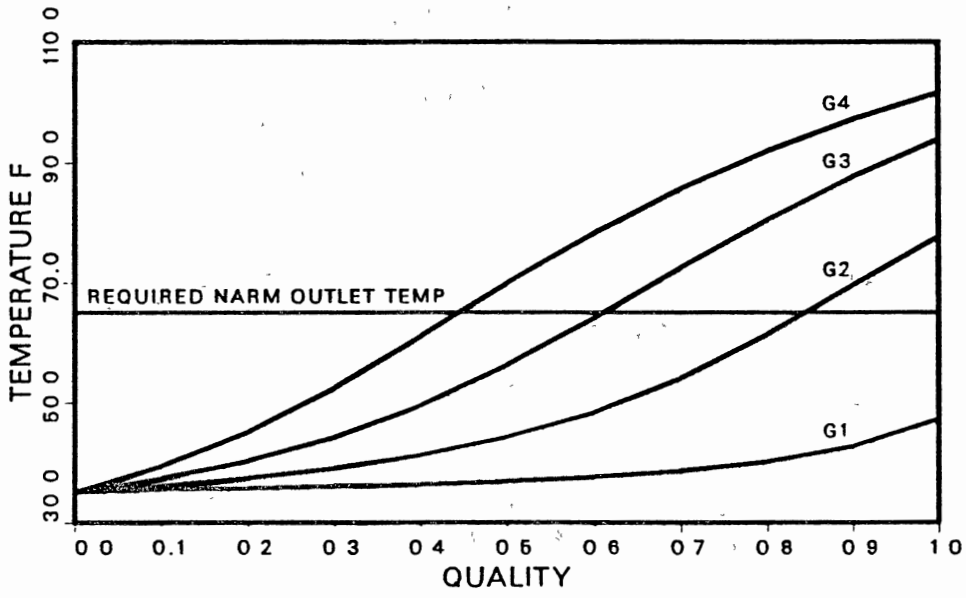


Figure 28. G-H NARM Temperature v/s Quality at Working Pressure

mixture with composition between G3 and G4 can be selected as the working fluid for a Type II evaporator because the 0.5 quality gliding temperature of these mixtures is nearly equal to the temperature loss of the source side. NARM's with composition between G1 and G2 can be selected as the working fluid for a Type I evaporator. To eliminate NARM's which are not suitable for the cycles with Type II evaporator, a computer program has been written to select the possible non-azeotropic binary refrigerant mixtures for cycles operating at a specified pressure and for a specified temperature loss of the source side. 15°F and 30°F source temperature losses are included in this study. The result of the survey is given in Figure 29 for a 15°F of 0.5 quality gliding temperature, and in Figure 30 for a 30°F of 0.5 quality gliding temperature. The data in Figures 29 and 30 are shown the saturated pressure (P) of liquid NARM at 35°F and the mole fraction x of the constituent which indicated on the horizontal axis of these figures.

The results in Figures 24, 25, 29 and 30 are based on a saturated NARM liquid existing at the inlet of the evaporator. In actuality, a subcooled liquid NARM is flashed through an expansion device and a low quality two-phase NARM exists at the evaporator inlet. Furthermore a NARM has a nonlinear temperature/enthalpy characteristic which affects the performance of the system. Therefore, the above study is only a guideline for selecting the constituents and composition of NARM's. The results in Figures 24, 25, 29 and 30 will be used as starting points for the simulation models in order to determine optimal selection of refrigerant, compositions, and cycles.

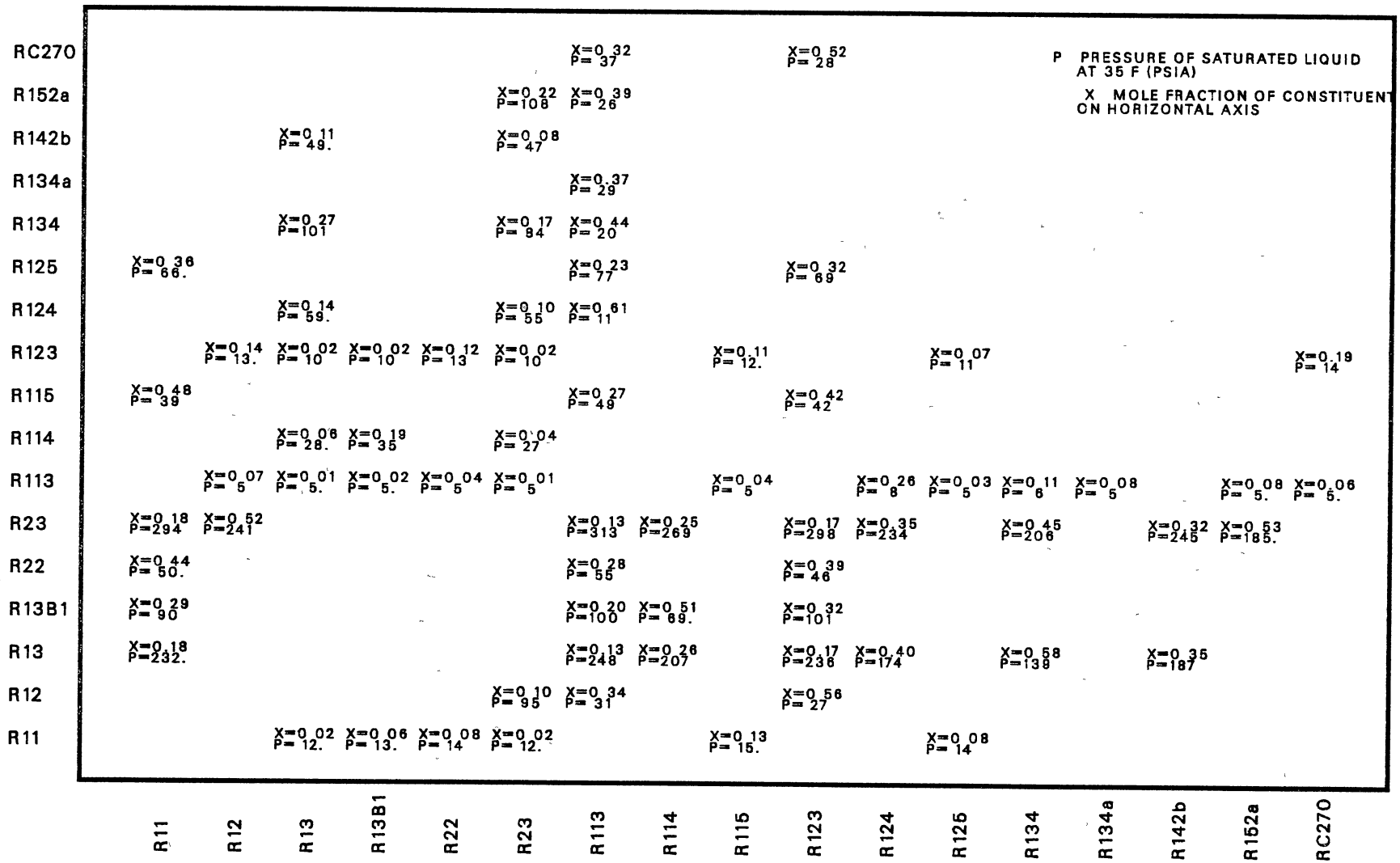


Figure 29. 35°F Sat. Liquid Pressure, Mole Fraction of binary Mixtures with 15°F Temperature Difference Between Sat. Liquid and 0.5 Quality

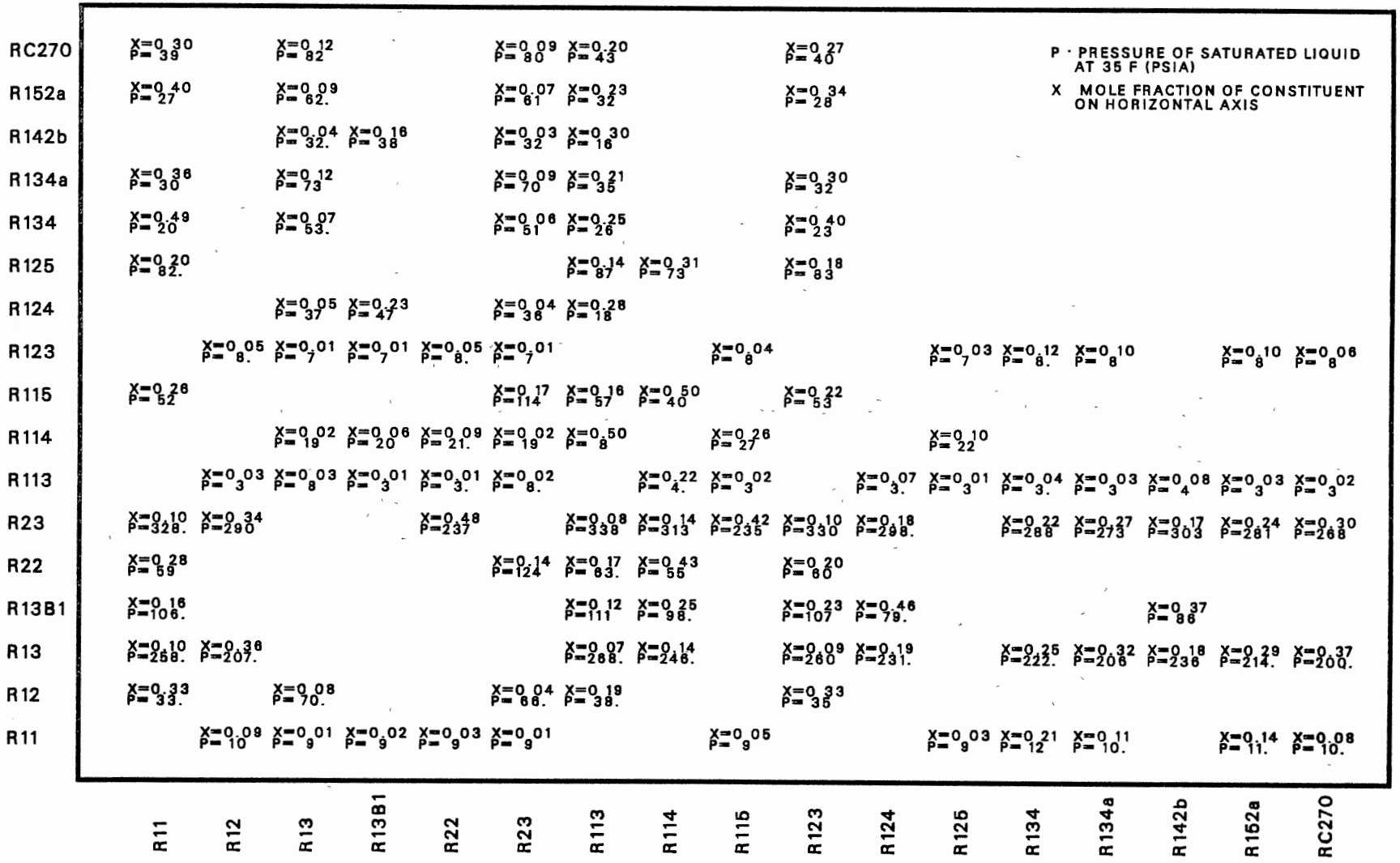


Figure 30. 35°F Sat. Liquid Pressure, Mole Fraction of Binary Mixtures with 30°F Temperature Difference Between Sat. Liquid and 0.5 Quality

CHAPTER IV

THERMODYNAMIC ANALYSIS AND SIMULATION TECHNIQUE

4.1 Component Thermodynamic Analysis

In this section, the thermodynamic characteristics of the components of vapor compression cycles will be analyzed.

The irreversibility (I) of a body is defined as the difference between the reversible work (ideal work) for a given change of state and the actual work that results in the same change of state.

$$I = W_{rev} - W_{c.v.} \tag{4.1}$$

Where W_{rev} : reversible work

$W_{c.v.}$: work across control volume

To estimate the irreversibility of the cycle, the irreversibility for individual components of the vapor compression cycles must first be analyzed. The continuity equation and the first and second law of thermodynamics are applied for each component. The symbols for the analysis are as follows:

m : Mass flow rate)

P : Absolute pressure

T : Temperature

h : Enthalpy

s : Entropy

x : Refrigerant mixture mole fraction

y : Quality

T_o : Surrounding absolute temperature

The subscripts 1,2,... are for state point 1,2,...

The subscripts h and c are for hot and cold stream of a heat exchanger, respectively.

The irreversibility analysis is based on References (1) and (59)

4.1.1 Expansion Device

In order to control the flow of liquid refrigerant between the low and high side of a vapor compression cycle, some form of an expansion device must be used. Most of existing expansion devices are designed for pure refrigerant systems. These devices need to be modified to operate with NARM systems. In this study, only thermodynamic characteristics of expansion device are emphasized. The three main expansion and control devices used to obtain automatic control for a pure refrigerant system are:

- Automatic expansion valve
- Thermostatic expansion valves
- Capillary tubes.

Automatic expansion valve. The function of an automatic expansion valve is to maintain a constant pressure in the evaporator. These valves are usually employed on a direct-expansion refrigerator because the system will usually operate in the constant heat sink and heat source temperature mode. Automatic expansion valves are not satisfactory in air conditioning due to the heat sink temperature and load fluctuations (31),(64).

Thermostatic expansion valve (TXV). The function of a thermostatic expansion valve is to regulate the flow of refrigerant into the evaporator in exact proportion to the rate of evaporation of the

refrigerant in the evaporator, to achieve a specified amount of superheat thus preventing the return of liquid refrigerant to the compressor.

For a pure refrigerant cycle, the thermostatic expansion valve is a hermetically sealed device that regulates the refrigerant to the evaporator using a thermal sensing element to monitor superheat. The saturation properties of the fluid in the bulb (and a spring preload) determine the evaporator pressure, thus control the evaporator saturation temperature.

Thermal expansion valves are satisfactory for air conditioning and heat pumps which operate with load and heat sink temperature fluctuations (31),(64).

Capillary tubes. A capillary-tube consists of a tube with an extremely small bore and a very long length. The reduction in pressure from the condenser to the evaporator is not achieved with a valve, but by the pressure drop through the tube. The subcooled liquid is flashed in the capillary tube.

Because of the gliding temperature of the NARM in the evaporator, all the above devices need to be modified to operate with a NARM vapor compression cycle. For a required fixed degree of superheat at the evaporator, an electronic TXV may be installed. An electronic TXV uses a microprocessor to store values of the superheated NARM vapor temperature as a function of pressure for a required fixed degree of superheat. Using a sensing element to measure temperature and pressure at the required point (e.g., the inlet to the compressor) to regulate the TXV opening and therefore control the flow rate through the evaporator.

4.1.1.a Thermodynamic and Irreversible Analysis

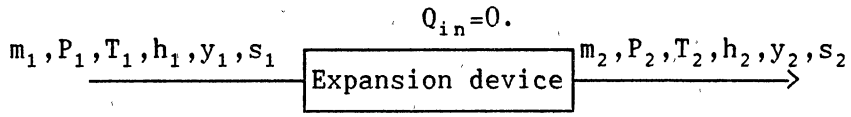


Figure 31. The Throttling Process

The throttling process is shown schematically in Figure 31. The continuity equation gives:

$$m = (m_1) = (m_2) \tag{4.2}$$

First law of thermodynamics: (A steady-state, steady-flow process)

$$(m_1 h_1) = (m_2 h_2) \tag{4.3}$$

$$\text{or } h_1 = h_2 \tag{4.4}$$

Second law of thermodynamics:

$$(m_2 s_2) \geq (m_1 s_1) \tag{4.5}$$

$$\text{or } s_2 \geq s_1 \tag{4.6}$$

Reversible work of the throttling valve W_{rev} for the change of state from state (1) to state (2)is found by:

$$(W_{rev}) = m\{(h_1-h_2) - T_o(s_1-s_2)\} \tag{4.7}$$

$$\text{or } (W_{rev}) = mT_o(s_2-s_1) \tag{4.8}$$

Work across control volume $W_{c.v.}$ for the throttling valve is equal to zero.

Thus the irreversibility of expansion device is:

$$I = (W_{rev})-0. = mT_o(s_2-s_1) \tag{4.9}$$

4.1.1.b Expansion Device Model

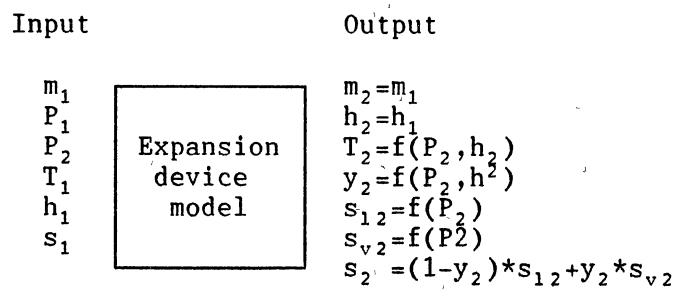


Figure 32. Expansion Device Model

The expansion device is modeled as an isenthalpic process and the model is shown in Figure 32

4.1.2 Liquid Pump

A liquid pump is employed in Cycle 6 to pump the low pressure subcooled liquid to high pressure. Addition of the liquid pump to the system results in increased total weight, and system cost. However, the total power consumption of the system may be reduced because the power consumption for a pump is much less than for a compressor. Furthermore, the liquid pump can be used in a "free cooling" mode. Free cooling mode occurs when the heat sink temperature is lower than the heat source temperature. For this mode, only the liquid pump needs to circulate refrigerant through the condenser and the evaporator. Because the compressor is turned off in a free cooling mode, the system power

consumption in a free cooling mode is significant less than during a normal mode. The liquid pump is modeled with an isentropic efficiency η_p and an electric motor efficiency η.

4.1.2.a Irreversibility Analysis

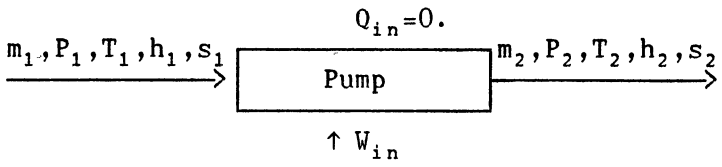


Figure 33. Pumping Process

The pumping process is shown in Figure 33. The continuity equation gives:

$$m = (m_1) = (m_2) \tag{4.10}$$

First law of thermodynamics: (A steady-state, steady-flow process)

$$(m_1 h_1) = (m_2 h_2) + W_{in} \tag{4.11}$$

$$\text{or } mh_1 = mh_2 + W_{in} \tag{4.12}$$

Second law of thermodynamics: the process is adiabatic for the control volume around the liquid pump

$$(m_2 s_2) \geq (m_1 s_1) \tag{4.13}$$

$$\text{or } s_2 \geq s_1 \tag{4.14}$$

Reversible work of the pump W_{rev} for the change of state .

$$(W_{rev}) = m\{(h_1 - h_2) - T_o(s_1 - s_2)\} \tag{4.15}$$

From Equation (4.12), the work across control volume $W_{c.v.}$ for the pump is:

$$W_{in} = m(h_1-h_2) \tag{4.16}$$

Thus, from Equation (4.15) and Equation (4.16), the irreversibility of the liquid pump is:

$$I = (W_{rev}) - W_{in} = mT_o(s_2-s_1) \tag{4.17}$$

4.1.2.b Thermodynamic Model

Liquid pump model is shown in Figure 34. A subcooled liquid in the pump at state 1 at pressure P_1, temperature T_1, enthalpy h_1 and entropy s_1, the isentropic process yields:

$$s_{2i} = s_1 \tag{4.18}$$

With the outlet pressure P_2 and entropy s_{2i} at state 2i, the enthalpy is then obtained from working fluid properties data.

From the definition of isentropic efficiency, η_p for a pump:

$$\eta_p = (h_{2i}-h_1)/(h_2-h_1) \tag{4.19}$$

The pump outlet enthalpy, h_2 is obtained by:

$$h_2 = \eta_p(h_{2i}-h_1)+h_1 \tag{4.20}$$

With the outlet pressure P_2, and enthalpy h_2, then temperature T_2, and entropy s_2 of the subcooled liquid at the pump outlet are obtained from working fluid properties. The work input to the pump is given by:

$$W_{in} = m_1(h_2-h_1) \tag{4.21}$$

Considering the pump electric motor efficiency, η_m, the electric power input to the pump, W_{ep}, is obtained from

$$W_{ep} = W_{in}/\eta_m \tag{4.22}$$

The power consumed by the pump will be included in the total power consumption to calculate the system performance.

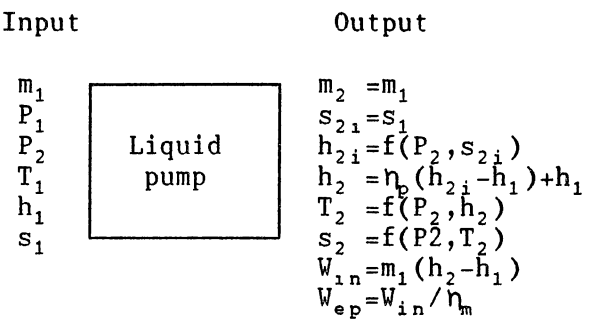


Figure 34. Liquid Pump Model

4.1.3 Separator

Cycle 6 needs a separator to separate the two-phase NARM into a saturated liquid and saturated vapor at different compositions. The regular receiver can be used as a separator. Because of the different densities, NARM vapors and NARM liquids can be separated by the gravity force and then a subcooled NARM liquid is pumped from the bottom of the receiver, and saturated vapor is drawn out at the top of the receiver. The separating process is shown in Figure 35. The continuity equation gives:

$$m_i = m_1 + m_2 \tag{4.23}$$

First law of thermodynamics: (A steady-state, steady-flow process)

$$(mh)_i = (mh)_1 + (mh)_2 \tag{4.24}$$

$$\text{let} \quad x = \frac{m_1}{m_1 + m_2} = \frac{m_1}{m_i} \tag{4.25}$$

Dividing Eq. (4.24) by m_i gives $h_i = xh_1 + (1.-x)h_2$ (4.26)

Second law of thermodynamics: $(ms)_1 + (ms)_2 \geq (ms)_i$ (4.27)

$$\text{or} \quad xs_1 + (1.-x)s_2 \geq s_i \tag{4.28}$$

Reversible work of the separator W_{rev} for the change of state is:

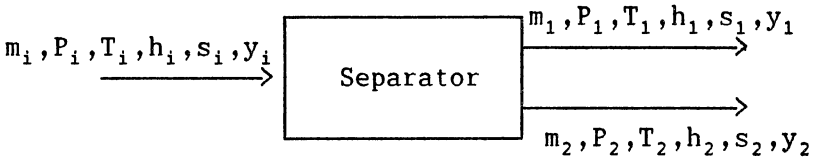


Figure 35. Separating Process

$$(W_{rev}) = \{m(h-T_o s)\}_i - \{m(h-T_o s)\}_1 - \{m(h-T_o s)\}_2 \quad (4.29)$$

or $$= m_i\{x(h-T_o s)_1 + (1-x)(h-T_o s)_2 - (h-T_o s)_i\} \quad (4.30)$$

$$= m_i T_o(s_1+s_2-s_e) \quad (4.31)$$

Work across the control volume $W_{c.v.}$ for the separator is zero. Therefore, the irreversibility is:

$$I = (W_{rev}) - 0. = m_i T_o(s_1+s_2-s_i) \quad (4.32)$$

4.1.3.b Thermodynamics Model

Separator model is shown in Figure 36. In this study, the pressure drop in the separator is assumed to be zero and the separator efficiency is 1.0.

The refrigerant is in equilibrium. Thus,

$$P_1 = P_2 = P_i \quad (4.33)$$

$$T_1 = T_2 = T_i \quad (4.34)$$

From the inlet quality y_i, the outlet saturated vapor flow rate m_1 is given by

$$m_1 = m_i y_i \quad (4.35)$$

The outlet saturated liquid flowrate m_2 is given by

$$m_2 = m_i(1.-y_i) \quad (4.36)$$

From the composition x_i, the temperature T_i, and the pressure P_i at the inlet of the separator, the composition of the saturated vapor

x_1 and saturated liquid x^2 are obtained from the mixture properties. The enthalpy h_1 and entropy s_1 of the saturated vapor are obtained from the working fluid properties for composition x_1 at temperature T_1 and pressure P_1. Similarly, the enthalpy h_2 and entropy s_2 of the saturated liquid are obtained from the working fluid properties for composition x_2 at temperature T^2 and pressure P^2.

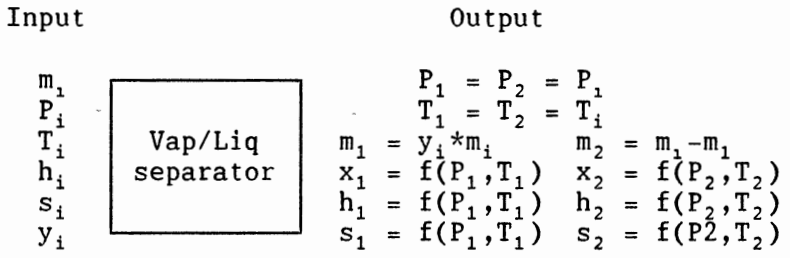


Figure 36. Separator Model

4.1.4 Adiabatic Mixing

The mixing process of two flows is shown in Figure 37. The mixing of two dissimilar streams of fluids may differ with regard to the thermodynamic states, phases or chemical composition. An example is the mixing of two liquid refrigerants at the same pressure but not at the same temperature. The continuity equation gives:

$$m_1 + m_2 = m_e \tag{4.37}$$

First law of thermodynamics: (A steady-state, steady-flow process)

$$(mh)_1 + (mh)_2 = (mh)_e \tag{4.38}$$

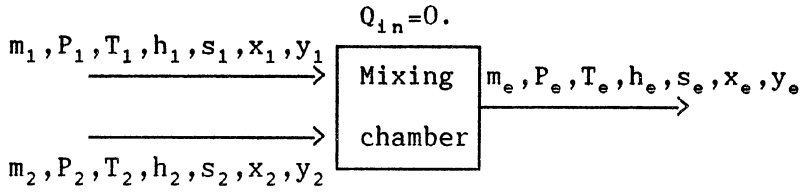


Figure 37. Mixing Process

$$\text{let} \quad x = \frac{m_1}{m_1 + m_2} = \frac{m_1}{m_e} \tag{4.39}$$

Dividing Eq. (4.38) by m_e gives $xh_1 + (1.-x)h_2 = h_e$ (4.40)

Second law of thermodynamics: the process is adiabatic for the control volume around the mixing chamber

$$(ms)e \geq (ms)_1 + (ms)_2 \tag{4.41}$$

$$\text{or} \quad s_e \geq xs_1 + (1.-x)s_2 \tag{4.42}$$

The reversible work of the mixing flow W_{rev} for the change of state is:

$$(W_{rev}) = \{m(h-T_o s)\}_1 + \{m(h-T_o s)\}_2 - \{m(h-T_o s)\}_e \tag{4.43}$$

$$\text{or} \quad = m_e\{x(h-T_o s)_1 + (1-x)(h-T_o s)_2 - (h-T_o s)_e\} \tag{4.44}$$

$$= m_e T_o(s_e - s_1 - s_2) \tag{4.45}$$

The work across the control volume $W_{c.v.}$ for the mixing chamber is zero. Therefore, the irreversibility of the adiabatic mixing is:

$$I = (W_{rev}) - 0. = m_e T_o(s_e - s_1 - s_2) \tag{4.46}$$

4.1.4.b Thermodynamic Model

Two binary streams at the same pressure with different concentration, enthalpy and temperature and/or different phases mix in a steady flow process is shown in Figure 38.

The concentration x_e and the enthalpy h_e of a mixing flow are:

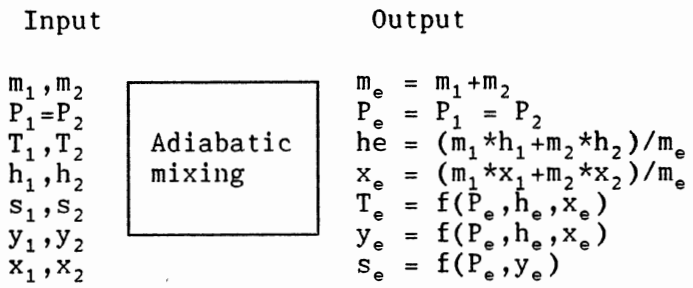


Figure 38. Adiabatic Mixing Model

$$x_e = x_1 + (m_1/m_e)(x_2 - x_1) \tag{4.47}$$

$$h_e = h_1 + (m_1/m_e)(h_2 - h_1) \tag{4.48}$$

Assuming the inlet pressure $P_1 = P_2$, then the mixing pressure is

$$P_e = P_1 = P_2 \tag{4.49}$$

The temperature T_e and the entropy s_e of the mixing flow are then obtained from the working fluid properties concentration x_e at the pressure P_e and the enthalpy h_e.

4.1.5 Compressor Model

4.1.5.a Irreversibility Analysis

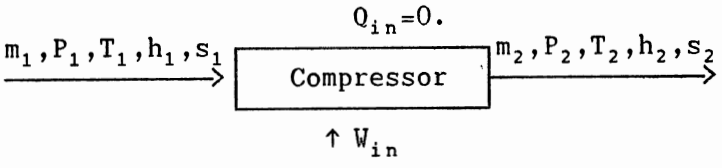


Figure 39. Compressing Process

Compressing process is shown in Figure 39. The continuity equation gives:

$$m = (m_1) = (m_2) \tag{4.50}$$

First law of thermodynamics: (A steady-state, steady-flow process)

$$(m_1 h_1) = (m_2 h_2) + W_{in} \tag{4.51}$$

$$\text{or } mh_1 = mh_2 + W_{in} \tag{4.52}$$

Second law of thermodynamics: the process is adiabatic for the control volume around the compressor.

$$(m_2 s_2) \geq (m_1 s_1) \tag{4.53}$$

$$\text{or } s_2 \geq s_1 \tag{4.54}$$

Reversible work of the compressor W_{rev} for the change of state is

$$(W_{rev}) = m\{(h_1-h_2) - T_o(s_1-s_2)\} \tag{4.55}$$

From Equation (4.52), the work across the control volume $W_{c.v.}$ for the compressor is $W_{in} = m(h_1-h_2)$ (4.56)

Thus, from Equation (4.55) and Equation (4.56) the irreversibility of the compressor is:

$$I = (W_{rev}) - W_{in} = mT_o(s_2-s_1) \tag{4.57}$$

4.1.5.b Thermodynamic Model

The compressor was modeled for simulating a non-isentropic process with isentropic efficiency η_c. The compressor model is shown in Figure 40. For superheated vapor at state 1 at, pressure P_1, temperature T_1, enthalpy h_1 and entropy s_1, the isentropic process gives

$$s_{2i} = s_1 \tag{4.58}$$

With the discharge pressure P_2 and entropy s_{2i}, at state 2i the enthalpy is then read from working fluid properties data.

From the definition of isentropic efficiency, η_c, of a compressor:

$$\eta_c = (h_{2i}-h_1)/(h_2-h_1) \tag{4.59}$$

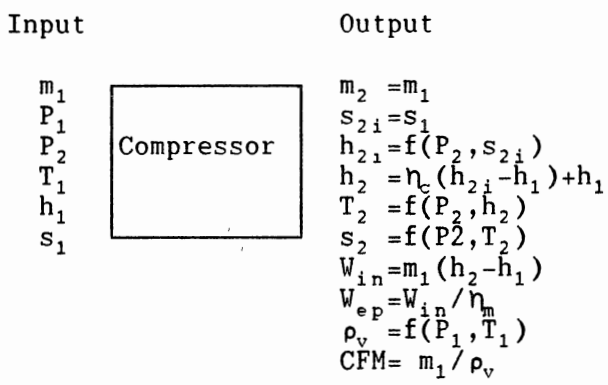


Figure 40. Compressor Model

The discharge enthalpy, h_2, is obtained from:

$$h_2 = h_1 + (h_{2i} - h_1)/\eta_c \tag{4.60}$$

Using the discharge pressure, enthalpy, and working fluid properties data, the temperature, enthalpy and entropy at the discharge are defined. Therefore, the work input to the compressor is:

$$W_{in} = m_1(h_2 - h_1) \tag{4.61}$$

With the compressor electric motor efficiency η_m, the power consumption input to the compressor W_{ec} is obtained from

$$W_{ec} = W_{in}/\eta_m \tag{4.62}$$

Suction volumetric flow rate (CFM) at state point 1 is calculated based on the mass flowrate and specific volume of refrigerant prior the compression process.

4.1.6 Motor Cooling

The compressor motor is assumed to be cooled and lubricated by the high pressure two-phase or subcooled liquid refrigerant. The maximum

outlet quality of refrigerant is assumed to be 1.0. The compressor motor losses are included in the estimated COP of the system.

4.1.6.a Irreversibility Analysis

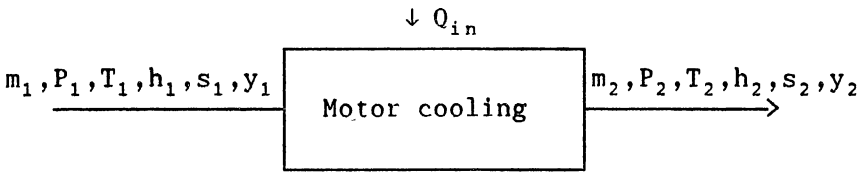


Figure 41. Motor Cooling Process

For the compressor and motor cooling considered together, the heat transfer rate to the surroundings is equal to zero. Q_{in} is considered to be the internal heat transfer rate from the electrical and mechanical losses to the cooling flow. The motor cooling process is shown in Figure 41. The reversible work, W_{rev}, for the change from state 1 to state 2 is:

$$W_{rev} = mT_o[(s_2-s_1)-(h_1-h_2)/T) \quad (4.63)$$

$$W_{in} = 0. \quad (4.64)$$

Therefore, the cooling flow irreversibility is

$$I = mT_o[(s_2-s_1)-(h_1-h_2)/T) \quad (4.65)$$

4.1.6.b Thermodynamic Model

The compressor motor losses, Q_{in}, are determined by

$$Q_{in} = m_{comp}*(h_{out}-h_{in})*(1.-\eta_m)/\eta_m \qquad (4.66)$$

where m_{comp}: Compressor flow rate

h_{in} : Compressor inlet enthalpy

h_{out} : Compressor outlet enthalpy

η_m : Electric motor efficiency

The heat transfer rate to the motor cooling flow is equal the losses in the compressor motor ($Q_{in} = Q_1$.). Therefore, the outlet enthalpy is obtained from :

$$h_2 = h_1 + Q_{in}/m \qquad (4.67)$$

Assuming the pressure drop of cooling flow is DP, the outlet pressure is :

$$P_2 = P_1 - DP \qquad (4.68)$$

At the outlet pressure P_2, and enthalpy h_2, the temperature T_2, quality y_2 and entropy s_2 at the outlet of the motor cooling flow is obtained from the refrigerant properties. The motor cooling model is shown in Figure 42.

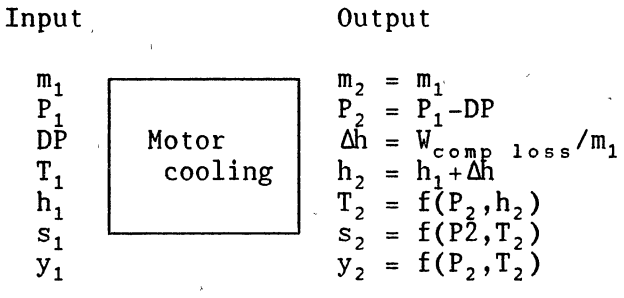


Figure 42. Motor Cooling Model.

4.1.7 Heat Exchanger Model

The evaporator, condenser, and subcoolers are modeled as counter flow heat exchangers.

4.1.7.a Irreversibility Analysis

Heat exchangers involve heat transfer with bodies at a temperature above or below the temperature of the surroundings. The irreversibility of the heat exchanger is obtained by:

$$I = \Sigma\, m_e T_o s_e - \Sigma\, m_i T_o s_i - Q_{cv}. \qquad (4.69)$$

With the assumption of no heat transfer from the heat exchanger to the surroundings, $Q_{cv} = 0$.. Therefore,

$$I = T_o*\{[m(s_e-s_i)]_h+[m(s_e-s_i)]_c\} \qquad (4.70)$$

For the condenser and evaporator, the change of the entropy between the outlet and the inlet of the single phase side is estimated as follows:

- for the liquid heat source or heat sink, the entropy is obtained from property data.

- for the air heat source or heat sink, the entropy difference is determined by:

$$\Delta s = C_p \ln (T_e/T_i) \qquad (4.71)$$

4.1.7.b Thermodynamic Models

- Evaporator model. The evaporating process in the evaporator is shown in Figure 43. The evaporator model requires the refrigerant inlet conditions.

For a NARM working fluid: pressure P_1, temperature T_1.

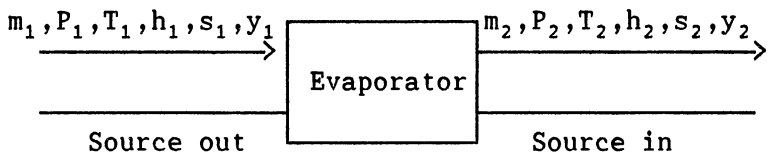


Figure 43. Evaporating Process

For a pure refrigerant: temperature T_1, enthalpy h_1.

The enthalpy h_1, the entropy s_1 and the quality y_1 at the inlet to the evaporator are obtained from the working fluid properties. The refrigerant outlet pressure is $P_2 = P_1 - DP$ (4.72)

where DP is the refrigerant pressure drop in the evaporator.

The evaporator model is shown in Figure 44. The model requires the refrigerant outlet quality y_2 (e.g., $y_2 \geq 1.$ for the cycles required saturated or superheated vapor exits from the evaporator (evaporator type I) or $y_2 < 1.$ for the cycles required two-phase refrigerant exits from the evaporator (evaporator type II) .

At the exit pressure P_2 and quality y_2, the exit temperature T_2, enthalpy h_2 and entropy s_2 are obtained from the working fluid properties.

The flow rate of the working fluid is obtained from:

$$m_1 = Q_{in}/(h_2-h_1) \quad (4.73)$$

where Q_{in} is the design heat load.

- Condenser Model. The condensing process in the condenser is shown in Figure 45. The condenser requires the refrigerant pressure P_2 and temperature T_2 at the outlet of the condenser for a NARM working

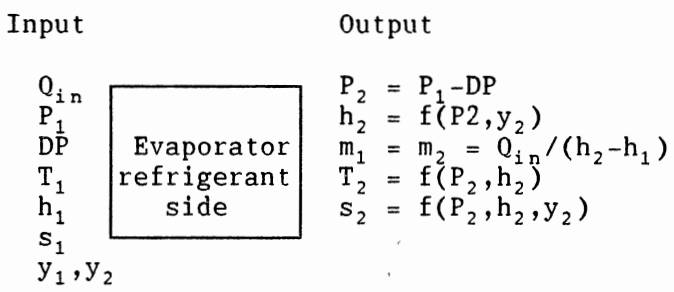


Figure 44. Evaporator Model

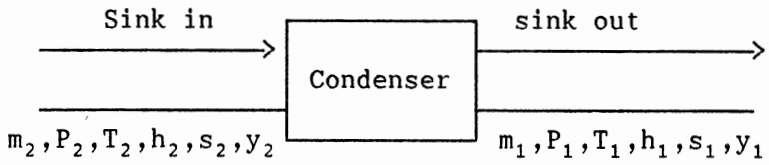


Figure 45. Condensing Process

fluid. For pure refrigerants, the temperature T_2 and the degree of subcooling at the condenser outlet must be known. The enthalpy h_2, the entropy s_2 and the quality y_2 at the outlet of the condenser are obtained from the working fluid properties. The refrigerant inlet pressure is defined by $P_1 = P_2 + DP$ (4.74)

where DP is the pressure drop of refrigerant in the condenser.

The inlet conditions of the condenser is defined by the flowrate m_1, the temperature T^1, the enthalpy h^1, entropy s^1, quality y^1 and composition x_1 are calculated from the previous steps of the cycle simulation model. Therefore, the heat rejected in the condenser is:

$$Q_{re} = m_1 * (h_1 - h_2) \quad (4.75)$$

The condenser model is shown in Figure 46.

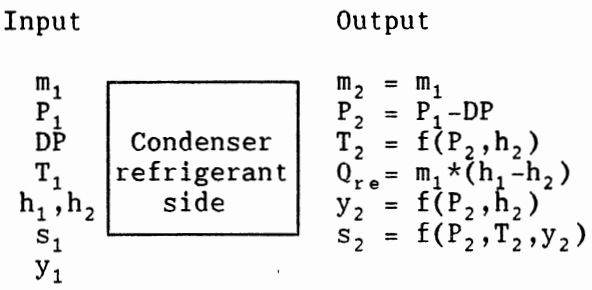


Figure 46. Condenser Model

- Subcooler Model. The subcooler is used in refrigeration cycles to reduce the temperature of the high pressure hot refrigerant by evaporating the low pressure cold refrigerant. Depending on the cycle, the high temperature refrigerant may be subcooled liquid or two-phase refrigerant. The subcooling process is shown in Figure 47.

The subcooler model requires the inlet conditions (state point 1 and 3), the pressure, temperature or enthalpy, as well as the pressure drop for the hot and cold sides of the heat exchanger. In addition, the model requires the outlet conditions of one side to be either hot or cold side (point 2 or 4). The subcooler model is shown in Figure 48.

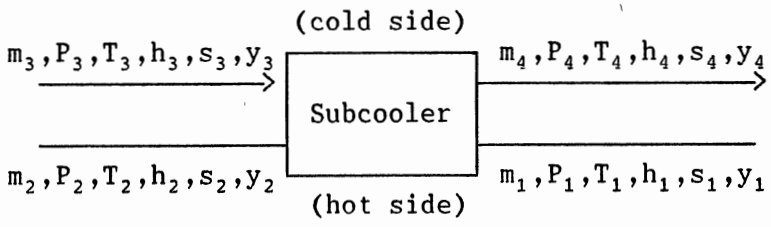


Figure 47. Subcooling Process

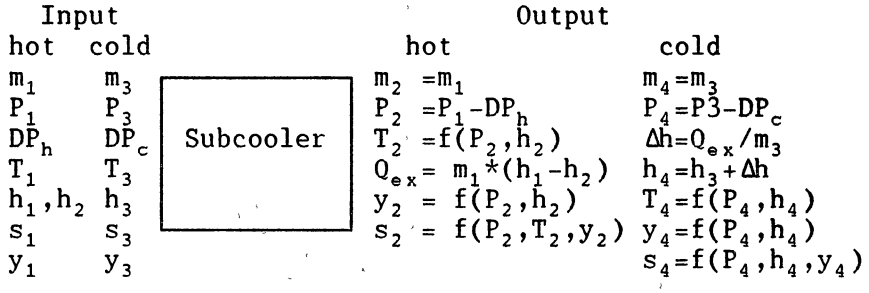


Figure 48. Subcooler Model

- Model to check the pinch point temperature and estimate the conductance of heat exchanger. Many NARM's have a nonlinear temperature-enthalpy behavior in the two phase region. The shape of the curves in this type of diagram can have a large influence on the effective mean temperature difference in the heat exchanger for a given effectiveness. Under certain conditions, a pinch point may occur which limits the potential of decreasing the temperature differences at the inlet or outlet of an evaporator or condenser. Or given an effectiveness, the pinch point will require an increased heat transfer area to achieve a desire heat load. Under other conditions, avoiding pinch points will result in a more desirable thermal performance of the heat exchanger. The performance of the heat exchanger is specified in terms of an average effective temperature difference and refrigerant pressure drop. The conductance value (UA) of the heat exchanger will allow a preliminary determination to be made concerning the size of the heat exchangers required.

In the calculation the conductance value, UA, of the heat exchangers, the vapor and liquid phases are assumed to be in

equilibrium with one another at any flow cross-section of the heat exchanger and the heat transfer coefficient is constant. The above assumptions represent an approximation because many factors affect the heat transfer coefficient as indicated in Reference (4). These factors may be two-phase quality, flow regime, the phase maldistribution, the mass transfer, and mass flowrate maldistribution of two-phase refrigerant to the channels of evaporator, etc. However, it becomes too complicated to consider the above effects in this study.

The conductance UA of each heat exchanger is calculated by dividing the heat exchangers into a number of finite difference segments, each of equal size Δq as shown in Figure 49. The heat exchanger conductance is then obtained by summing the quantity of the heat transferred over each segment divided by log mean temperature difference at that segment as shown below.

Calculate the temperature difference between the hot side and the cold side at segment i, $\Delta T_i = T_{hi} - T_{ci}$ (4.76)

Calculate the temperature difference between the hot side and the cold side at segment i+1

$$\Delta T_{i+1} = T_{hi+1} - T_{ci+1} \quad (4.77)$$

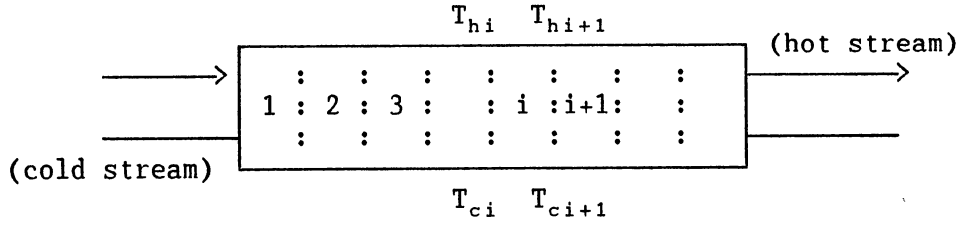


Figure 49. Heat Exchanger Model

The log mean temperature difference at segment i is obtained from

$$LMTD_i = (\Delta T_{i+1} - \Delta T_i)/Ln(\Delta T_{i+1}/\Delta T_i) \quad (4.78)$$

The conductance of heat exchanger is determined by:

$$UA = \Sigma\ (\Delta q/LMTD) \quad (4.79)$$

where the subscripts h and c refer to the hot and cold streams respectively.

The pressure drop for each fluid is assumed to be evenly distributed for each segment of the heat exchanger.

The heat exchanger model computes the difference of temperature between two fluids, conductance and the pinch point temperature of the heat exchanger. If required, a temperature profile will be drawn.

4.1.8 Condenser Fan

For applications such as air conditioning or heat pumps, the power consumption for the condenser fan is an important factor in total power consumption of the system. Even liquid-cooled condenser chillers require power for the cooling tower fan.

In this study, the power consumption is estimated for one ton cooling load. The inlet and outlet temperatures of the source side are considered constant. As the result of the fixed cooling load and source temperature, the power required to pump heat source fluid (liquid or air) through the evaporator is the same for every cycle and for any refrigerant. Therefore, the power consumption for the evaporator fan or pump is not included in this study.

For the condenser, only the inlet sink temperature is constant. The outlet temperature of the sink side depends on required heat rejection in the condenser which depends on the performance of vapor compression cycle and the temperature of the refrigerant in the

condenser. Therefore, the sink flow rate can vary to meet the pinch point temperature requirement. The power consumption for the condenser fan or pump is varied for each cycle and the working fluid. The power consumption W_{fan} for the condenser fan is estimated by

$$W_{fan} = \frac{m_{air} * DP_{air}}{\rho_{air} * \eta_{fan}} \qquad (4.80)$$

where :

DP_{air} : air side pressure drop

ρ_{air} : air density

η_{fan} : fan efficiency (include electric motor)

4.1.9 Cycle

The irreversibility of the cycle is the sum of the irreversibility of the components. The irreversibility of the components is calculated based on the dead state temperature of 537°R (298°C) and air is heat sink and heat source. Summary of component irreversibility is given in Table IV.

The irreversibility of the cycle is obtained from:

$$I_{cycle} = \sum_{j=1,n} I_j \qquad (4.81)$$

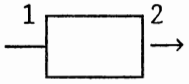
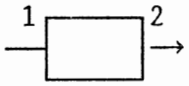
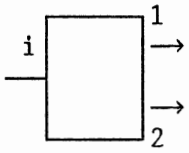
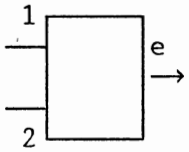
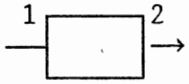
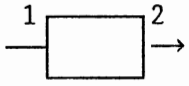
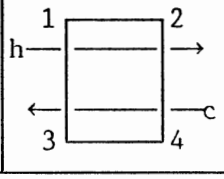
4.2 Cycle Thermodynamic Analysis

The performance of a refrigerating system is expressed in terms of the coefficient of performance (COP), which is defined as:

$$COP = \frac{\text{useful refrigerating effect}}{\text{Net energy input}} \qquad (4.82)$$

TABLE IV

SUMMARY OF COMPONENT IRREVERSIBILITY ANALYSIS

COMPONENT	SCHEMATIC	IRREVERSIBILITY
Expansion device		$mT_o(s_2-s_1)$
Liquid Pump		$mT_o(s_2-s_1)$
Separator		$m_i T_o(s_1+s_2-s_i)$
Adiabatic mixing		$m_e T_o(s_e-s_2-s_1)$
Compressor		$mT_o(s_2-s_1)$
Motor cooling		$mT_o(s_2-s_1-(h_1-h_2)/T)$
Condenser Evaporator Subcooler		$T_o\{m_h(s_2-s_1)+m_c(s3-s4)\}$

The following cycle performances are evaluated for a unit mass flow rate of refrigerant flow through the evaporator. For simplicity, pressure drops in the heat exchangers and connecting piping are neglected .

4.2.1 Cycle 1 and Cycle 3

The flow diagram with the state points are given in Figures 14a and 16a, and the P-H diagrams are given in Figures 14b and 16b. Refrigerant entering the compressor is saturated or slightly superheated vapor at the evaporator pressure. The compression process 1-22 is assumed to be non-isentropic with isentropic efficiency η_c, and is continued until the condenser pressure is reached. Point 22 is a superheated vapor. The process 22-4 is at nearly constant pressure, with the temperature of the superheated vapor decreasing until the saturated vapor condition is reached. The process is in both constant pressure and constant temperature for a pure refrigerant or constant pressure process for a NARM. The working fluid leaves the condenser at point 4 as a saturated or subcooled liquid. It is then expanded through a throttling valve where partial evaporation occurs as the pressure drops across the valve. The throttling process 4-5 is irreversible with an increase in entropy. The enthalpy at the exit and inlet valves are equal. A part of the liquid from the outlet of the condenser point 4 is used to cool the compressor drive motor before it is mixed with the existing slightly superheated vapor from the evaporator outlet and drawn into the compressor inlet to complete the cycle.

The useful refrigerating effect is :

$$Q_{evap} = Q_{5-1} = h_1 - h_5 \qquad (4.83)$$

Net energy input to the compressor

$$W_{comp} = W_{1-22} = h_{22} - h_1 \qquad (4.84)$$

Therefore the coefficient of performance is :

$$COP = \frac{Q_{evap}}{W_{comp}} = \frac{h_1 - h_5}{h_{22} - h_1} \qquad (4.85)$$

4.2.2 Cycle 2

The description of Cycle 2 is given in Chapter 2, the flow diagram with the state points are given in Figure 15a, and the P-H diagrams are given in Figures 15b. Furthermore, the two phase cooling stream leaving the subcooler HX (state point 20) at the interstage pressure level is used to cool the compressor drive motor before it is mixed with the first stage exiting vapor stream (state point 2).

The useful refrigerating effect is :

$$Q_{evap} = Q_{5-1} = h_1 - h_5 \qquad (4.86)$$

The net energy input to the first and second stage of the compressor is:

$$W_{comp} = W_{1-2} + W_{21-22} \qquad (4.87)$$

Where $W_{1-2} = h_2 - h_1$

$W_{21-22} = m_c(h_{22} - h_{21})$

m_c : refrigerant flow rate through the condenser

Therefore

$$COP = \frac{h_1 - h_5}{h_2 - h_1 + m_c(h_{22} - h_{21})} \qquad (4.88)$$

An energy balance for the subcooler HX yields

$$(h_4 - h_7) = (m_c - 1.)(h_{20} - h_{18}) \qquad (4.89)$$

At points 3,18, and 24 $h_3 = h_{18} = h_{24}$. Substitute to Equation (4.88) yields

$$(h_4 - h_7) = (m_c - 1.)(h_{20} - h_4) \qquad (4.90)$$

An energy balance for mixing of flow 2 and flow 21 yields:

$$m_c h_{21} = h_2 + (m_c - 1.) h_{20} \tag{4.91}$$

Solving Equations (4.88) and (4.89) for the mass flow rate m_c

$$m_c = \frac{(h_2 - h_7)}{(h_{21} - h_4)} \tag{4.92}$$

Substituting m_c into Equation (4.86). The coefficient of performance is:

$$COP = \frac{(h_1 - h_5)}{(h_2 - h_1) + \dfrac{(h_{22} - h_{21})(h_2 - h_5)}{(h_{21} - h_4)}} \tag{4.93}$$

Without the subcooler HX, h_5 approaches h_4, h_{21} approaches h_2, and the COP of Cycle 2 approaches the COP of Cycle 1.

4.2.3 Cycle 4

To get the benefit of NARM's, Cycle 4, Cycle 5 and Cycle 6 need a new component, a heat exchanger called "SUB 2". Cycle 4 is used to explain the basic function of the heat exchanger "SUB 2".

Figure 17a shows a flow diagram of a vapor compression Cycle 4, and the P-h diagrams are given in Figures 17b. In addition to the evaporator and the condenser, this cycle uses two more heat exchangers, SUB 1 and SUB 2.

The thermodynamic configuration of the vapor cycle was specifically designed for a system using a NARM. Liquid refrigerant leaving the condenser at point 4 enters subcooler I where it is further subcooled by evaporating a two-phase refrigerant at approximately the interstage pressure. Because the NARM evaporates with a rising temperature, the flow for this two phase cooling is tapped from the

subcooled liquid leaving the subcooler SUB 1 at point 7. Refrigerant then passes through an expansion device to the cold side of the subcooler SUB 1 which is a counterflow heat exchanger configuration. The two phase NARM leaves the subcooler SUB 1 at point 20 and provides further two phase cooling for the compressor motor before being injected into the superheated vapor leaving the first compressor stage at point 2. This interstage cooling increases the efficiency of the overall compression process and reduces the second stage outlet vapor temperature to within the safe operating limits for the refrigerants used in the NARM.

The hot subcooled liquid leaving subcooler SUB 1 at point 7 enters the subcooler SUB 2 for further subcooling, and is then routed to the evaporator. A small portion of subcooled refrigerant liquid leaving the subcooler SUB 2 is flashed through an expansion device and used as a cooling source for two phase cooling in subcooler SUB 2. The outlet of subcooler SUB 2 at point 26 is merged with the outlet of the evaporator point 1 to provide a slightly superheated vapor to the inlet of the compressor first stage at point 11. It should be noted that the subcooling done in the subcooler SUB 2 is only advantageous to a NARM system, allowing a reduction in the evaporator minimum temperature without a reduction in evaporator pressure. This is due to the slope of the constant temperature line in the two phase region of the NARM. This allows a lower overall compressor pressure ratio and may get higher cycle efficiency.

The exchanger, SUB 2, is a new element in the system, and it is like another evaporator in parallel with the "main" evaporator. It provides subcooling of the condenser liquid from T_7 to T_6.

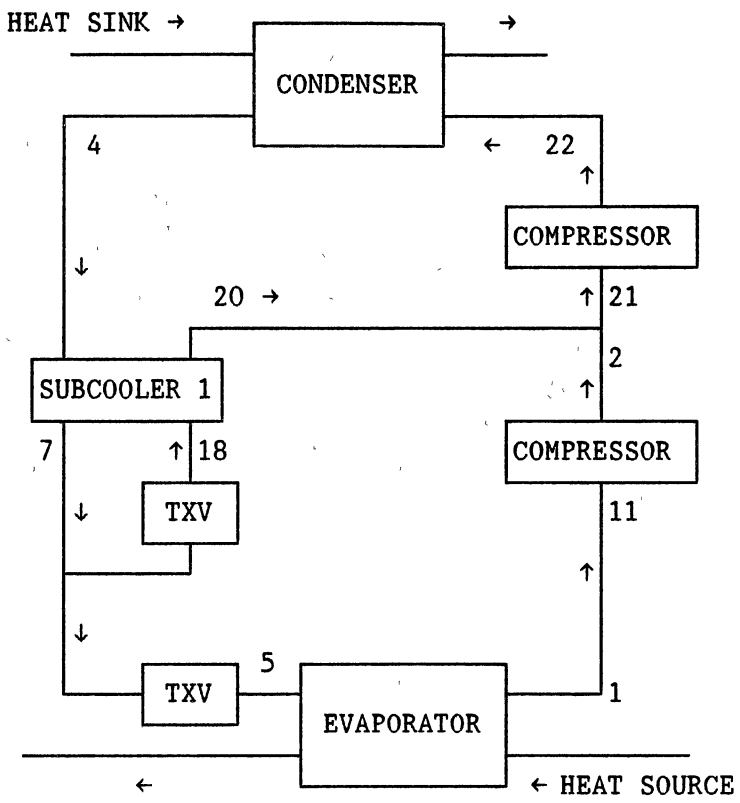


Figure 50. Cycle 4 without Subcooler SUB 2

The heat exchanger, SUB 1, is used to reduce the compressor work input and the maximum cycle temperature (at the compressor exit). To understand the effect of the subcooler SUB 2 in the cycle 4, the performance of cycle 4 without the heat exchanger SUB 2 must be calculated. The flow diagram and state point is given in Figure 50.

The useful refrigerating effect is :

$$Q_{evap} = Q_{5-1} = h_1 - h_5 \qquad (4.94)$$

The net energy input to first and second stage of the compressor is:

$$W_{comp} = W_{1-2} + W_{21-22} \qquad (4.95)$$

Where $W_{1-2} = h_2 - h_1$ (4.96)

$W_{21-22} = m_c(h_{22} - h_{21})$ (4.97)

m_c : refrigerant flow rate through the condenser

Therefore

$$COP = \frac{h_1 - h_5}{h_2 - h_1 + m_c(h_{22} - h_{21})} \quad (4.98)$$

An energy balance for subcooler SUB 1 gives

$$(h_4 - h_7) = (m_c - 1.)(h_{20} - h_{18}) \quad (4.99)$$

Through the expansion device, $h_7 = h_{18}$. Substituting into Equation (4.99) yields

$$(h_4 - h_7) = (m_c - 1.)(h_{20} - h_4) \quad (4.100)$$

An energy balance for the mixing of flow 2 and flow 21 yields:

$$m_c h_{21} = h_2 + (m_c - 1.)h_{20} \quad (4.101)$$

Solving Equations (4.99) and (4.100), for the mass flow rate

$$m_c = \frac{(h_2 - h_7)}{(h_{21} - h_4)} \quad (4.102)$$

Substituting m_c into Equation (4.98), the coefficient of performance of Cycle 4 without the subcooler SUB 2 is:

$$COP = \frac{(h_1 - h_7)}{(h_2 - h_1) + \frac{h_{22} - h_{21})(h_2 - h_5)}{(h_{21} - h_4)}} \quad (4.103)$$

Now, the performance of Cycle 4 is calculated including the subcooler SUB 2

Let m_{c1} is the total flow rate of NARM hot side to the heat exchanger SUB 2 (state 7) or the flow rate to the first stage of the compressor.

An energy balance in heat exchanger SUB 2 yields:

$$m_{c1}(h_7-h_6) = (m_{c1}-1.)(h_1-h_5) \qquad (4.104)$$

across the throttling valve 6-5

$$h_6=h_5 \ ; \quad T_7>T_1 \text{ and } T_6>T_5$$

Thus,

$$m_{c1} = (h_1-h_5)/(h_1-h_7) \qquad (4.105)$$

The energy balance for the heat exchanger SUB 1 yields

$$m_c(h_4-h_7) = (m_c-m_{c1})(h_{20}-h_{18}) \qquad (4.106)$$

across the throttle valve 7-18 yields $h_7=h_{18}$. Substituting into Equation (4.106)

$$m_c(h_4-h_7) = (m_c-m_{c1})(h_{20}-h_7) \qquad (4.107)$$

The energy balance for the mixing of flow 2 and flow 21 yields:

$$m_c h_{21} = m_{c1}h_2+(m_c-m_{c1})h_{20} \qquad (4.108)$$

Solve Equations (4.107) and (4.108) provides

$$\frac{m_c}{m_{c1}} = \frac{(h_2 - h_7)}{(h_{21} - h_4)}$$

$$m_c = \frac{(h_2-h_7)(h_1-h_5)}{(h_{21}-h_4)(h_1-h_7)} \qquad (4.109)$$

The COP of the system is

$$COP = \frac{(h_1-h_5)}{m_{c1}(h_2-h_1)+m_c(h_{22}-h_{21})} \qquad (4.110)$$

Substituting m_{c1} from Equation (4.105) and m_c from (4.109), the coefficient of performance of the cycle with the subcooler SUB 2 is given as

$$COP = \frac{(h_1-h_7)}{(h_2-h_1)+\dfrac{(h_{22}-h_{21})(h_2-h_5)}{(h_{21}-h_4)}} \qquad (4.111)$$

Note that the exchanger SUB 2 affects the cycle in the following ways :

(a) The COP (Equation (4.103) for Cycle 4 without the subcooler SUB2 and Equation (4.111) for Cycle 4 with the subcooler SUB2) does not directly depend upon exchanger SUB 2 .

(b) Figure 51 shows the cycle on a pressure-enthalpy (P-h) diagram. The saturation dome of the P-h diagram is qualitatively the same regardless of whether the working fluid is a single refrigerant or a binary mixture of a constant composition. As shown in this figure, the distinguishing feature is the fact that $T_5=T_8=T_1$ for a single refrigerant while $T_5<T_8<T_1$ for a binary nonazeotropic refrigerant mixture. Without the exchanger SUB 2, The fluid enters the evaporator at T_8 rather than at T_5. Since $T_5=T_8=T_1$ for a single fluid, the exchanger SUB 2 has no effect upon the pressure ratio and the refrigerant temperature in the evaporator. In summary, heat exchanger SUB 2 is useless in a system using a single refrigerant.

(c) For a NARM with $T_5<T_8<T_1$, the exchanger SUB 2 allows a reduction in temperature (from T_8 to T_5) at the evaporator entry for the same pressure ratio and COP as cycle without the heat exchanger. Alternatively, a decrease in the pressure ratio and increase in the COP are obtained with the use of exchanger SUB 2 for an evaporator inlet temperature of T_8. Thus the exchanger SUB 2 is essential for an efficient or effective use of a NARM.

For a pure refrigerant, heat is rejected from the cycle in the two-phase region at a constant temperature (T4). For a NARM, this process occurs over a temperature range from T4 to T22, which provides the benefit of a reduced cooling flow rate or decrease in condenser size.

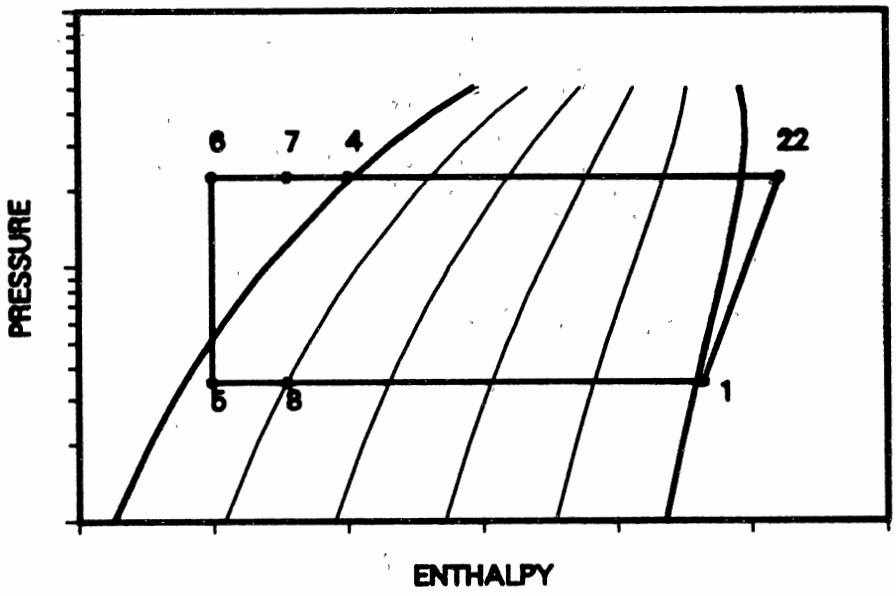


Figure 51. P-h Diagram

4.2.4 Cycle 5

The description of Cycle 5 is given in Chapter II. Figure 18a shows the flow diagram for Cycle 5. This cycle uses two heat exchangers, SUB 1 and SUB 2. SUB 1 is used to further condense the high pressure two-phase refrigerant leaving the condenser (state point 4). This is accomplished by evaporating the two-phase refrigerant leaving the evaporator (state point 2) to a higher quality. The two-phase refrigerant leaving the subcooler SUB 1 at point 8 is used to cool the compressor drive motor prior entry into the compressor at point 21. This exchanger allows an increase in the specific cooling effect (i.e. cooling effect per unit mass flow of the refrigerant). But the same

result can be achieved if the superheating of vapor within the evaporator is permissible. The adverse effect of exchanger I is an increase in the maximum cycle temperature T_{22}. Similar to cycle 4, the exchanger SUB 2 is a new element in the system, and it is like any other evaporator configured in parallel with the "main" evaporator. The additional exchanger provides subcooling of the condenser liquid from T_7 to T^6.

Assuming steady-state flow and neglecting the potential and the kinetic energy, an energy balance on heat exchanger SUB I yields:

$$m_c(h_{21}-h_1) = m_c(h_4-h_7) \text{ or}$$

$$(h_{21}-h_1) = (h_4-h_7) \; ; \; T_4>T_{21} \; , \; T_7>T_1 \qquad (4.112)$$

For heat exchanger SUB 2

$$m_c(h_7-h_6) = (m_c-1.)(h_1-h_5) \qquad (4.113)$$

Across the throttling valves,

$$h_6=h_5 \; ; \; T_7>T_1 \text{ and } T_6>T_5 \qquad (4.114)$$

Thus,

$$m_c= (h_1-h_5)/(h_1-h_7) \qquad (4.115)$$

The compressor work input, Win, is given by

$$Win = W_{21-22} = m_c(h_{22}-h_{21}) \qquad (4.116)$$

The cooling effect Qin is given by

$$Qin = (h_1-h_5) = (h_1-h_6) \qquad (4.117)$$

From Equations (4.116) and (4.117), the coefficient of the performance, COP, is given by

$$COP = \frac{Q_{evap}}{W_{comp}} = \frac{(h_1-h_5)}{\dfrac{(h_1-h_5)}{(h_1-h_7)}(h_{22}-h_{21})} \qquad (4.118)$$

or

$$COP = \frac{(h_1 - h_7)}{(h_{22} - h_{21})} \qquad (4.119)$$

The effects of the subcooler SUB 2 on Cycle 5 is similar to its effects on Cycle 4.

In this study, the compressor motor efficiency (η_m) is included to estimate the total power consumption for the vapor compression cycle.

4.3 Simulation Techniques

FORTRAN simulation models were constructed to predict the cycle performance using pure refrigerants or binary non-azeotropic refrigerant mixtures as the working fluids. The electrical power consumption is estimated for all electrical motors including the compressor, pump, and fan (except the pumping power required for the heat source). The compressor motor is assumed to be cooled and lubricated by the refrigerant and any losses are included in the estimated system COP. The program includes models for the expansion device, compressor, liquid pump, separator, condenser fan as well as the heat transfer processes in the condenser, evaporator, and subcoolers. Depending on the cycle, the refrigerant exits the evaporator as either a two-phase or super-heated vapor. The refrigerant exits the condenser as a two-phase mixture, saturated liquid or subcooled liquid. Two values of system performance are evaluated. The first for vapor compression cycle alone, while the second includes the condenser fan (or pump).

4.3.1 Overall Description of Cycles

The required input parameters for the simulation models are:

- Refrigerant
- HEATL : Heat load (Tons)
- T31 : Inlet heat source temperature (°C)
- T32 : Outlet heat source temperature (°C)
- T41 : Inlet heat sink temperature (°C)
- DPEVA : Evaporator pressure drop (kPa)
- DPC : Condenser pressure drop (kPa)
- DPCOOL : Cooling motor flow pressure drop (kPa)
- COMPEF : Compressor isentropic efficiency
- EFFMOT : Electric motor efficiency
- EFFFAN : Condenser fan efficiency
- DTSUP : Evaporator outlet degree superheat (°C)
- DTSUB : Condenser outlet degree subcool (°C)
- T5 : Evaporator inlet refrigerant temperature (°C)
- T4 : Condenser outlet refrigerant temperature (°C)
- NDIV : Number of segments of heat exchanger, use to check pinch point temperature
- DTPIN : Required minimum pinch point temperature (°C)
- DPSI : Condenser pressure drop for heat sink side (kPa)
- CPSO : Heat source specific heat (kJ/kg)
- CPSI : Heat sink specific heat (kJ/kg)
- IUNIT : The unit system for output print out (IUNIT= 1 for SI and = 0 for English system)

The output of the simulation model consist of data for all state points. The symbols are used in the simulation as follows:

FL : Flowrate (kg/min) or (lbm/min)

P : Pressure (kPa) or (psia)

T : Temperature (°C) or (°F)

H : Enthalpy (J/kg) or (Btu/lbm)

S : Entropy (J/kg-°C) or (Btu/lbm-°F)

Y : Quality

DEN : Density (kg/m_3) or (lbm/ft^3)

Data for the cycle:

PR : Compressor pressure ratio

VRATE : Inlet compressor volume flow rate (m^3) or (ft^3)

COP : Coefficient of performance (for VCS only)

COPT : System coefficient of performance including the power consumption for the compressor and the condenser fan (or pump)

UA : Conductance for all heat exchangers (kJ/min-°C) or (Btu/min-°F)

Irreversibility for all components: evaporator, condenser, expansion device, compressor, and motor cooling (kJ/min) or (Btu/min).

The symbols (FI_1,FI_2,...,FO_1,FO_2), to express refrigerant thermodynamic properties FO_1,FO_2 can be calculated from other known properties of refrigerants FI_1, and FI_2.

4.3.1.a Overall Description of the Cycle 1 Model

The flow schematic with the state points and P-h diagram in Figure 52 are used to explain the method used to predict the performance of Cycle 1.

The simulation starts from the point 4, with temperature T4 at the condenser outlet. With the condenser outlet degree of subcooling,

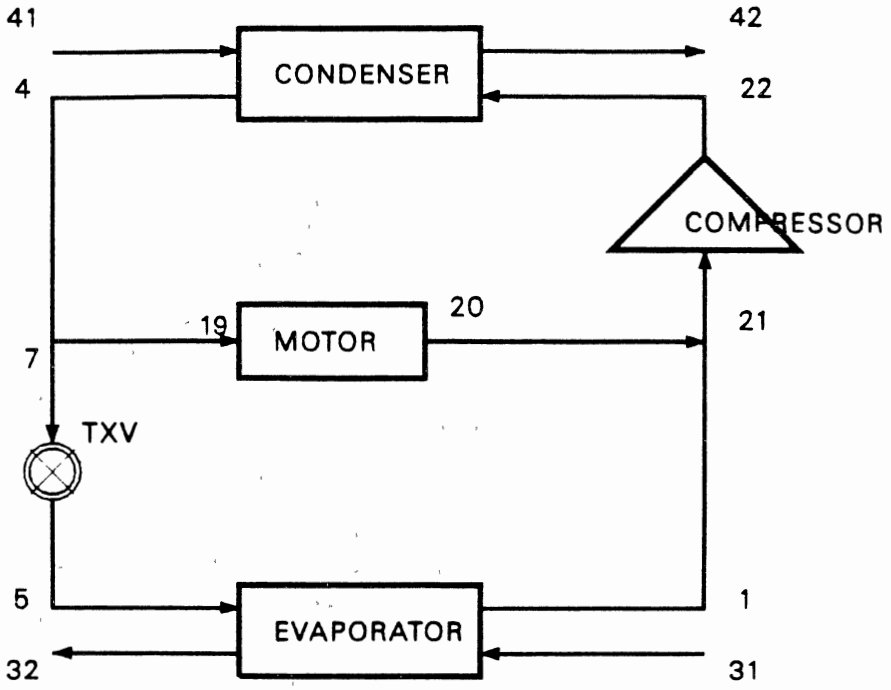


Figure 52a. Cycle 1 Flow Diagram and State Points

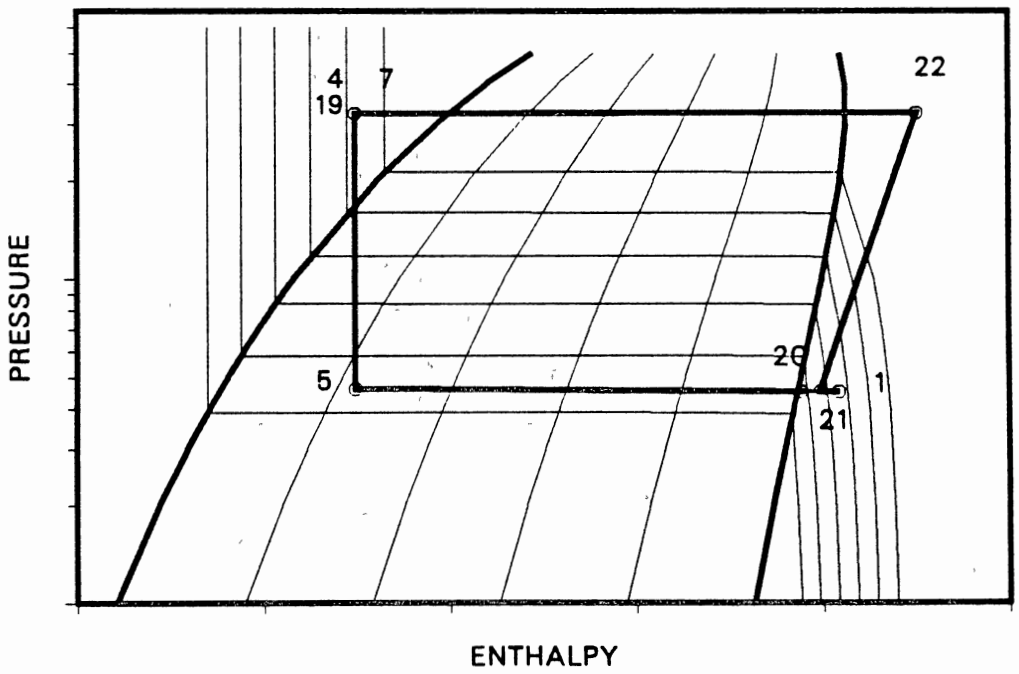


Figure 52b. Cycle 1 P-h Diagram

DTSUB, the refrigerant saturation temperature at point 4 is determined by

$$T4SAT = T4+DTSUB. \quad (4.120)$$

The pressure P4 is calculated from saturation properties of the refrigerant (T4,..,P4). The enthalpy, H4, of the subcooled liquid at point 4 is calculated from saturated liquid properties of the refrigerant (T4,..,H4). At this stage, all other refrigerant properties of state point 4 required for the simulation are calculated .

The enthalpy, H5, at point 5 is obtained by the process from point 4 through an isenthalpic expansion device. Thus, H5=H4. The pressure, P5, at point 5 is calculated from the saturated refrigerant properties at temperature T5 of the refrigerant entering the evaporator (T5,..,P5). At this point, all other refrigerant properties at state point 5 are calculated, including the quality Y5. With an evaporator pressure drop DPEVA, the refrigerant outlet pressure P1 is obtained from

$$P1 = P5-DPEVA \quad (4.121)$$

The saturation temperature at point 1 is then calculated from the refrigerant saturation properties (P1,..,T1SAT). Since the degree of superheat, DTSUP, is imposed at point 1, the temperature T1 is calculated by

$$T1 = T1SAT+DTSUP \quad (4.122)$$

All other properties of point 1 are obtained.

The refrigerant flowrate through the evaporator is

$$FL5=FL1=HEATL/(H1-H5) \quad (4.123)$$

The state at point 20 is assumed to be a saturated vapor with P20=P11=P1. H20 is calculated from the saturated vapor properties at P20 (P20,...,H20).

The simulation assumes a flow rate of FL20 at point 20 . The compressor inlet flowrate and enthalpy are given by:

FL21=FL1+FL20 and H21=(FL1*H1+FL20*H20)/FL21 (4.124)

With P21 and H21 determined, all other properties of superheated vapor at point 21 are obtained using the superheated vapor properties. With the condenser pressure drop DPCO, the pressure at the compressor discharge, point 22, is

P22 = P4+DPCO (4.125)

The isentropic compression process provides S22I=S21 and H22I is obtained from the refrigerant superheated vapor properties (P22,S22I,H22I). Then, an isentropic compressor efficiency COMPEF is taken into account to obtain the enthalpy at point 22

H22 = H11+(H22I-H11)*COMPEF (4.126)

The electric power supplied to the compressor is found from

POCOM = FL11*(H22-H21)/EFFMOT (4.127)

The motor loss to be cooled by the refrigerant at point 19 is :

POLOSS = POCOM(1.-EFFMOT) (4.128)

The enthalpy H19 at point 19 ,associated with the flowrate FL20 is obtained from

H19 = H20-POLOSS/FL20 (4.129)

If the energy balance of H19 = H4 is not satisfied, then the procedure is repeated with a new flowrate F20 until the condition is met. If they are within a specified limit, convergence is achieved, and the model calculates the other characteristics of the system. The compressor pressure ratio and suction volume flow rate is calculated from

PR = P22/P21 (4.130)

VRATE = FL21/DEN21 (4.131)

The flowrate of the heat source is

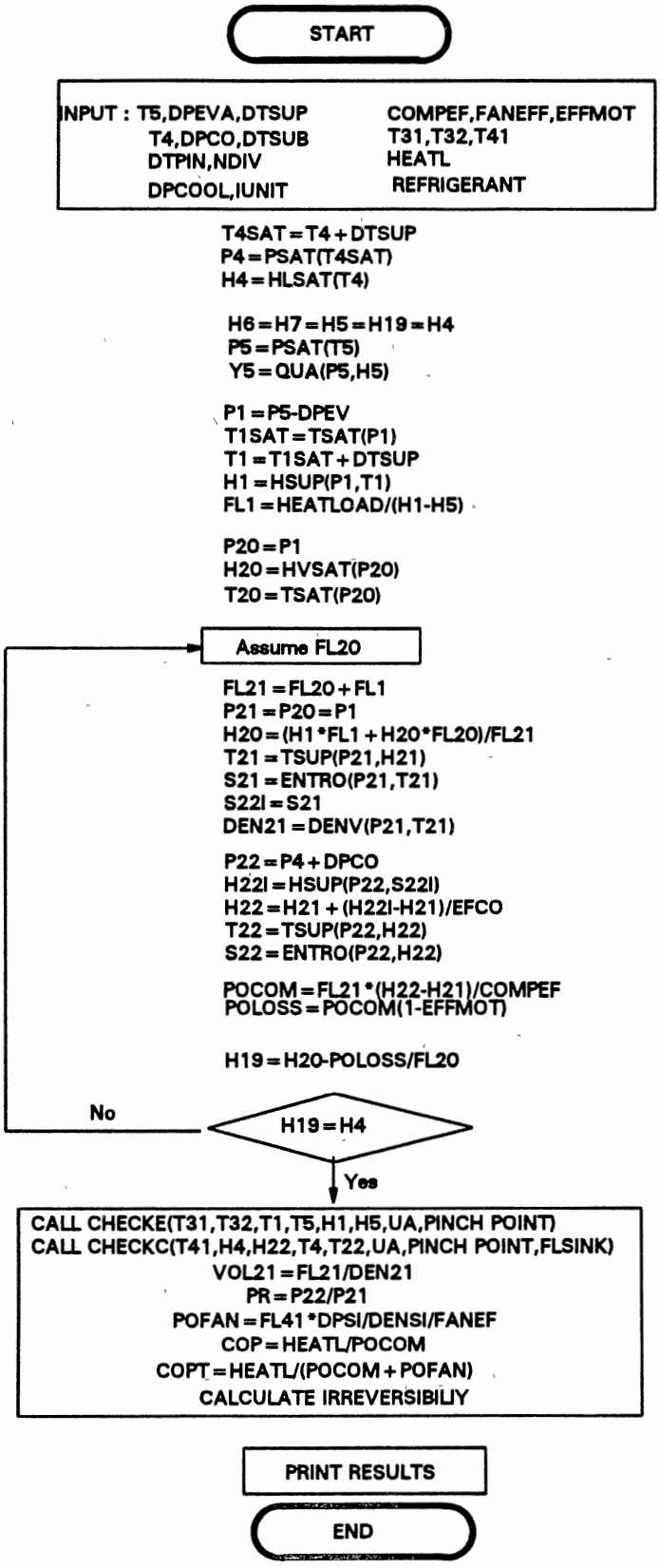


Figure 53. Flow Chart for Cycle 1

$$F31 = HEATL/(T31-T32)/CPsource \quad (4.132)$$

The CHECKE subroutine is called to check the pinch point temperature, and to calculate the conductance, UA, of the evaporator.

The simulation assumes a heat sink flowrate, F41. The CHECKC subroutine is called to calculate the pinch point temperature, DTPINC, and the conductance UA of the condenser. If the pinch point temperature requirement DTPINC≤DTPIN is not satisfied, then a new sink flowrate FL41 is updated until the condition is satisfied.

The power consumption for the condenser fan is found from

$$POFAN = FL41*DPSI/DENSI/FANEF \quad (4.133)$$

Then the coefficient of performances are obtained from:

$$COP = HEATL/POCOM \quad (4.134)$$

$$COPT = HEATL/(POCOM+POFAN) \quad (4.135)$$

The program then calculates the irreversibility of the components and system, and prints out the results. The summary flow chart of Cycle 1 is given in Figure 53.

4.3.1.b Overall Description of Cycle 2 Model

Besides the required input parameters for the Cycle 1, some additional parameters are required for the Cycle 2 as follows:

PR1 : First stage compressor pressure ratio

DTD : Temperature drop across expansion device (°C)

DPHXH : Subcooler hot side refrigerant pressure drop (kPa)

DPHXC : Subcooler cold side refrigerant pressure drop (kPa)

DTSUP2 : Compressor second stage inlet degree superheat (°C)

The flow schematic with the state points and P-h diagram in Figure 54 are used to explain the method to predict the performance of Cycle 2. The simulation starts from the point 4, with refrigerant at a

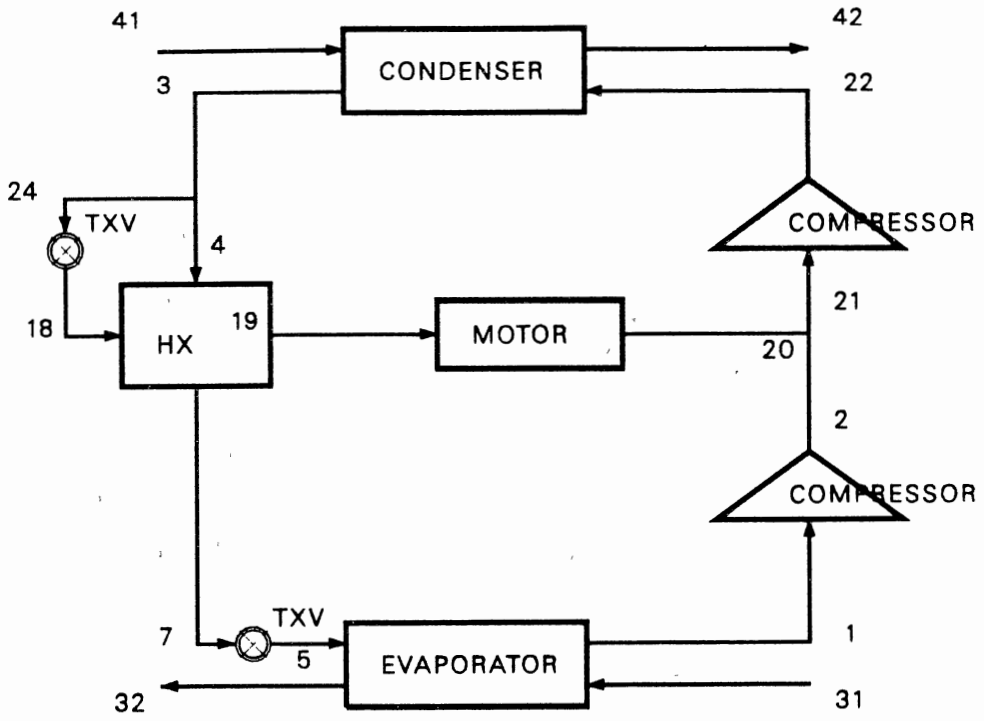


Figure 54a. Cycle 2 Flow Diagram and State Points

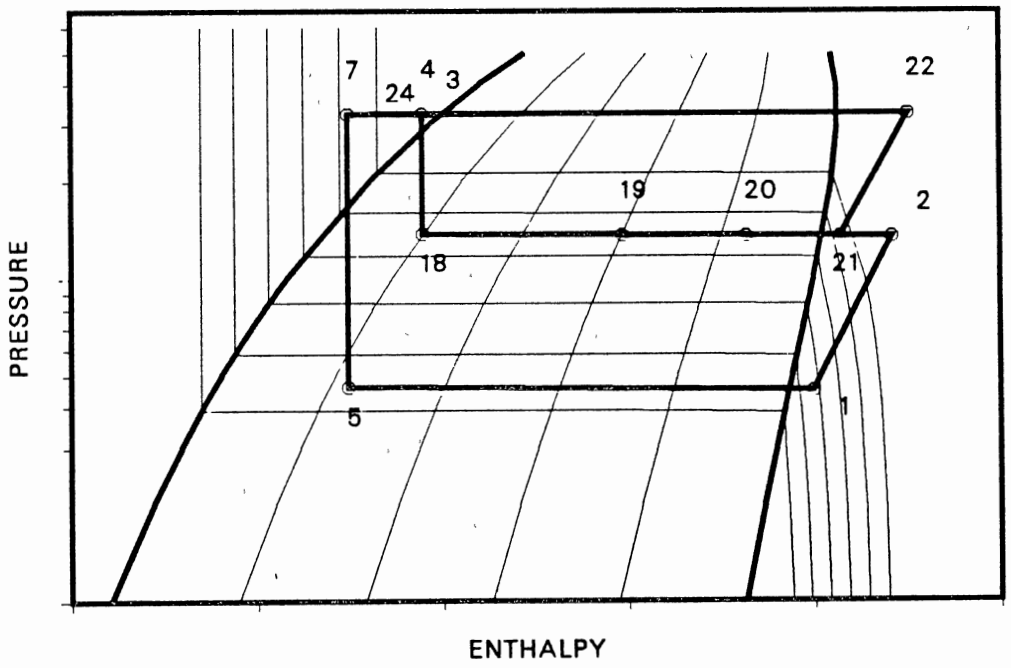


Figure 54b. Cycle 2 P-h Diagram

temperature T4. With the condenser outlet degree subcooling DTSUB, the refrigerant saturation at point 4 is determined by

$$T4_{sat} = T4+DTSUB \qquad (4.136)$$

The pressure P4 is calculated from saturation properties of the refrigerant (T4,...,P4). The enthalpy, H4, of the subcooled liquid at point 4 is calculated from saturation properties of refrigerant (T4,...,H4). At this stage all other properties of state point 4 required for the simulation are also calculated.

At point 7, the pressure P7 is calculated by

$$P7 = P4-DPHXH \qquad (4.137)$$

the temperature T7 is determined by

$$T7 = T5+DTD \qquad (4.138)$$

The enthalpy H7 is calculated from the saturated refrigerant liquid properties (T7,...,H7)

The enthalpy H5 at point 5 is obtained by the process from point 7 through an isenthalpic expansion device, therefore H5=H7. The pressure P5 at point 5 is calculated from refrigerant saturation properties at temperature T5 of refrigerant entering the evaporator (T5,...,P5). At this point, all other properties at state point 5 are calculated, including the quality Y5. With an evaporator pressure drop, DPEVA, the refrigerant outlet pressure P1 is obtained from

$$P1 = P5-DPEVA \qquad (4.139)$$

The saturation temperature at point 1 is calculated from the refrigerant saturation properties (P1,...,T1SAT). Since the degree of superheat DTSUP is imposed at point 1, the temperature T1 is calculated by

$$T1 = T1SAT+DTSUP \qquad (4.140)$$

At this stage all other properties of point 1 are obtained, including the evaporator outlet enthalpy H1, entropy S1, vapor density DEN1.

The refrigerant flowrate through the evaporator is

$$FL5 = FL1=HEATL/(H1-H5) \quad (4.141)$$

With the first stage compressor ratio PR1, the first stage compressor discharge pressure at point 2 is

$$P2 = P1*PR1 \quad (4.142)$$

The isentropic compression process yields S2I=S1 and H2I is calculated from the refrigerant superheated vapor properties (P2,S2I,...,H2I). With an isentropic compressor efficiency COMPEF, the enthalpy at point 2 is

$$H2 = H1+(H2I-H1)/COMPEF \quad (4.143)$$

Other properties at point 2 are also calculated at this point.

The electric power supplied for the first stage of the compressor is

$$POCOM1 = FL1*(H2-H1)/EFFMOT \quad (4.144)$$

The pressure at point 21 is

$$P21 = P20 = P2 \quad (4.145)$$

The saturation vapor temperature at point 20, T20SAT, is obtained from the saturated vapor properties at pressure P20. With the degree of superheat, DTSUP2, at the compressor second stage inlet, the temperature at point 21 is

$$T21 = T20SAT+DTSUP2 \quad (4.146)$$

With P21 and T21, all other properties of superheated vapor at point 21 are calculated from the superheated vapor properties.

The simulation assumes a flow rate of FL20 at point 20. The flowrate of the compressor second stage is given by:

$$FL21 = FL2+FL20 \quad (4.147)$$

An energy balance at point 20 gives

$$H20 = (FL21*H21-FL2*H2)/FL20 \quad (4.148)$$

With the condenser pressure drop, DPCO, the pressure at the second stage compressor discharge, point 22, is

$$P22 = P4+DPCO \quad (4.149)$$

The isentropic compression process give S22I=S21 and H22I is calculated from the refrigerant superheated vapor properties (P22,S22I,...,H22I). Then an isentropic compressor efficiency COMPEF is taken into account to obtain the enthalpy at point 22

$$H22 = H21+(H22I-H21)/COMPEF \quad (4.150)$$

The electric power supplied to the second stage of the compressor is obtained from $POCOM2 = FL21*(H22-H21)/EFFMOT$ (4.151)

Therefore, the total motor loss and need to be cooled by the refrigerant at point 19 is:

$$POLOSS = (POCOM1+PCOM2)(1.-EFFMOT) \quad (4.152)$$

At point 19, the enthalpy H19 and the pressure P19 is determined from

$$H19 = H20-POLOSS/FL20 \quad (4.153)$$

$$\text{and } P19 = P20-DPMOT \quad (4.154)$$

The enthalpy at point 18 is determined by an energy balance in the heat exchanger. Therefore, the enthalpy H18 associated with the assumed flowrate F20.

$$H18 = H19-FL1/FL20*(H4-H7) \quad (4.155)$$

If the energy balance of H18=H4 is not satisfied, the procedure is repeated with a new flowrate F20 until the condition is met. If they are within a specified limit, convergence is achieved and the model calculates the other characteristics of the system. The second stage compressor pressure ratio and the suction volume flow rates are calculated from

PR2 = P22/P21 (4.156)

VRATE1 = FL1/DEN1 (4.157)

VRATE2 = FL21/DEN21 (4.158)

The flowrate of the heat source is

FL31 = HEATL/(T31-T32)/CPSO (4.159)

The simulation assumes a heat sink flowrate, FL41. The CHECKC subroutine is called to calculate the pinch point temperature DTPINC and the conductance, UA, of the condenser. If the pinch point temperature requirement (DTPINC $\leq$ DTPIN) is not satisfied, the new sink flowrate FL41 is updated until the condition is satisfied. The model calculates the power consumption for the condenser fan from

POFAN = FL41*DPSI/DENSI/FANEF (4.160)

The CHECKE subroutine is called to check the pinch point temperature, and calculate the conductance, UA, of the evaporator.

The CHECKHX subroutine is called to check the pinch point temperature, and calculate the conductance, UA, of the heat exchanger.

The coefficients of performance are obtained from:

COP = HEATL/(POCOM1+PCOM2) (4.161)

COPT = HEATL/(POCOM1+POCOM2+POFAN) (4.162)

The model continues to calculate the irreversibility of components and the system, and prints out the results.

The summary flow chart for Cycle 2 is given in Figure 55.

4.3.1.c Overall Description of Cycle 5 Model

The flow schematic with the state points and P-h diagram in Figure 56 are used to explain the method to predict the performance of Cycle 5. Besides the input parameters required for Cycle 1, some additional parameters are required for the cycle 5 as follows:

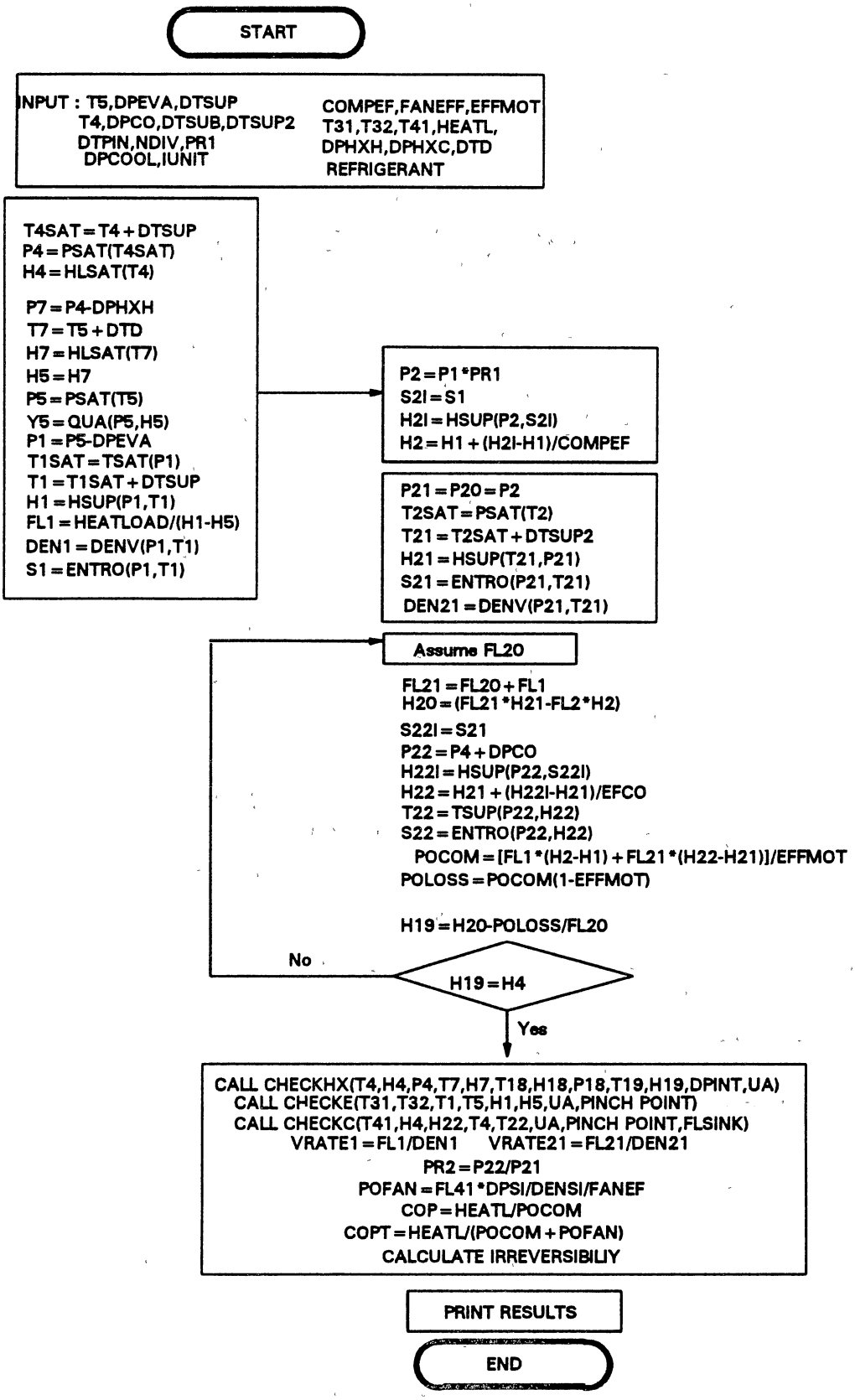


Figure 55. Flow Chart for Cycle 2

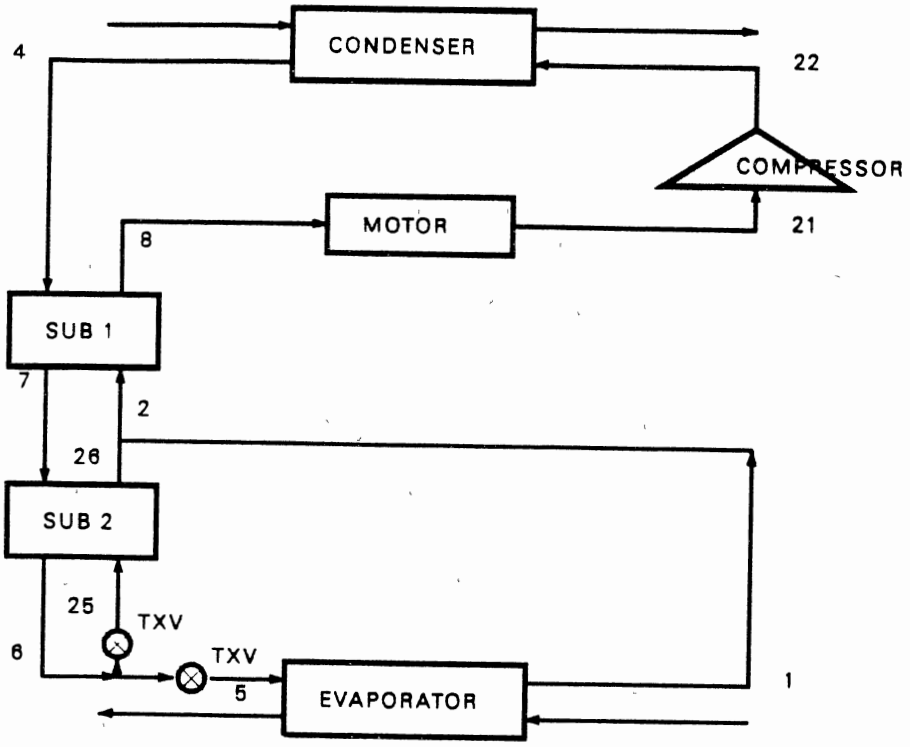


Figure 56a. Cycle 5 Flow Diagram and State Points

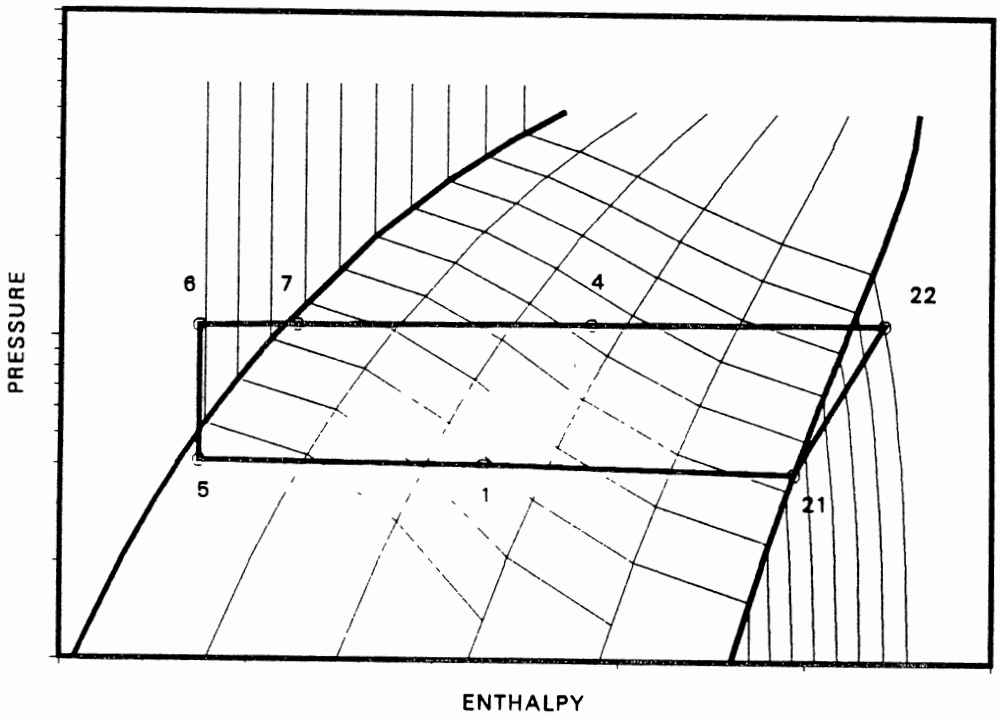


Figure 56b. Cycle 5 P-h Diagram

XMOL : Refrigerant mole fraction

DTD : Temperature drop across expansion device (°C)

DPHXH : Subcooler hot side refrigerant pressure drop (kPa)

DPHXC : Subcooler cold side refrigerant pressure drop (kPa)

DTSUP2 : Degree superheat at the compressor inlet (°C)

The simulation starts at point 6. The outlet temperature of the hot side of the subcooler II is obtained from

$$T6 = T5+DTD \tag{4.163}$$

The enthalpy H6 is calculated from the properties of the saturated NARM liquid.

The enthalpy H5 at point 5 is obtained by the process from point 6 through an isenthalpic expansion device, H5=H6. The pressure P5 and quality Y5 at point 5, is calculated from two-phase NARM properties at the temperature T5 and the enthalpy H5. At this stage in the process, all other properties at state point 5 are calculated. With an evaporator pressure drop DPEVA, the refrigerant outlet pressure P1 is obtained from

$$P1 = P5-DPEVA \tag{4.164}$$

The saturation temperature of saturation NARM vapor at point 1 is calculated from the properties of saturated NARM vapor. Since the degree of superheat DTSUP is imposed at point 1, the temperature T1 is calculated by

$$T1 = T1SAT+DTSUP \tag{4.165}$$

The enthalpy H1 and quality Y1 at point 1 is calculated from the two-phase NARM properties at temperature T1 and pressure P1. Note, the evaporator of the Cycle 5 is a Type II evaporator. In which, a two-phase refrigerant exits the evaporator. The outlet degree superheat,

DTSUP, is only a symbol and has a negative number in this case. The evaporator flowrate may then be determined from:

$$FL5 = FL1 = HEATL/(H1-H5) \quad (4.166)$$

The system is designed with points 2, 1 and 26 satisfied by

$$P2 = P26 = P1 \quad (4.167)$$

$$T2 = T26 = T1 \quad (4.168)$$

$$\text{and } H2 = H26 = H1 \quad (4.169)$$

With the subcooler II cold side pressure drop, DPHXC, the refrigerant pressure at point 8 is obtained from

$$P8 = P2-DPHXC \quad (4.170)$$

therefore, the pressure at point 21 is

$$P21 = P8-DPCOOL \quad (4.171)$$

The saturation temperature of the saturated NARM vapor T21SAT at point 21 is calculated from properties of a saturated NARM vapor. Since the degree of superheat DTSUP2 is imposed at point 21, the temperature T21 is calculated by

$$T21 = T21SAT+DTSUP2 \quad (4.172)$$

The enthalpy H21, entropy S21, and the vapor density DEN21 at point 21 are calculated from superheated vapor NARM properties at temperature T21 and pressure P21. With the compressor pressure ratio, PR, the compressor discharge pressure at point 22 is obtained from

$$P22 = P21*PR \quad (4.173)$$

The isentropic compression process gives

$$S22I = S21 \quad (4.174)$$

and H22I is calculated from the superheated vapor NARM properties. At this point, an isentropic compressor efficiency COMPEF is taken into account to obtain the enthalpy at point 22

$$H22 = H21+(H22I-H21)/COMPEF \quad (4.175)$$

The electric power supplied to the compressor for one unit of mass flowrate is found by

$$DELPO = (H22-H21)/EFFMOT \quad (4.176)$$

Therefore, the motor loss per unit of mass flowrate to be cooled by the refrigerant at point 8 is

$$DELH = (H22-H21)(1-EFFMOT)/EFFMOT \quad (4.177)$$

Then at point 8, the enthalpy H8 and the pressure P8 is determined from

$$H8 = H21-DELH \quad (4.178)$$

and

$$P8 = P21-DPCOOL \quad (4.179)$$

The condenser outlet pressure is

$$P4 = P22-DPCO \quad (4.180)$$

The enthalpy H4 and quality Y4 at point 4 are calculated from the two-phase NARM properties at temperature T4 and pressure P4.

At point 7, withe the subcooler I hot side pressure drop, DPHXH, the outlet pressure at the subcooler I hot side outlet is

$$P7 = P4-DPHXH \quad (4.181)$$

The enthalpy H7 is obtained from an energy balance on the subcooler I and is

$$H7 = H4-H8+H2 \quad (4.182)$$

The temperature T7 is defined from the NARM properties. The NARM flowrate through the condenser is determined from the energy balance of the subcooler II

$$FL4 = FL7 = FL2 = FL21 = FL5*(H6-H26)/(H7-H26) \quad (4.183)$$

At point 6, the outlet pressure of the the subcooler I hot side is

$$P6 = P7-DPHXH \quad (4.184)$$

The electric power supplied to the compressor is

$$POCOM = FL21*(H22-H21)/EFFMOT \quad (4.185)$$

The compressor suction volume flow rate is calculated from

$$VRATE = FL21/DEN21 \qquad (4.186)$$

The flowrate of the heat source is

$$FL31 = HEATL/(T31-T32)/CPSO \qquad (4.187)$$

The heat sink flowrate FL41 is assumed. The CHECKC subroutine is called to calculate the pinch point temperature, DTPINC, and the conductance, UA, of the condenser. If the pinch point temperature requirement (DTPINC $\leq$ DTPIN) is not satisfied, a new sink flowrate FL41 is assumed until the condition is satisfied. Then the model calculates the power consumption for the condenser fan

$$POFAN = FL41*DPSI/DENSI/FANEF \qquad (4.188)$$

The CHECKE subroutine is called to check the pinch point temperature and calculate the conductance UA of the evaporator.

The coefficients of performance are obtained from:

$$COP = HEATL/POCOM \qquad (4.189)$$

$$COPT = HEATL/(POCOM+POFAN) \qquad (4.190)$$

The CHECKHX subroutine is called to check the pinch point temperature and calculate the conductance, UA, of the subcooler I and subcooler II.

The program continues to calculate the irreversibility of the components and the system, and print out the results.

The summary flow chart of Cycle 5 is given in Figure 57

4.3.2 Assumptions

In order to fairly compare the performance of various fluids and cycles, it is reasonable to require that the cooling effect, Q_{evap}, is constant regardless of the working fluids and cycles. This is accomplished by specifying source stream temperatures entering and

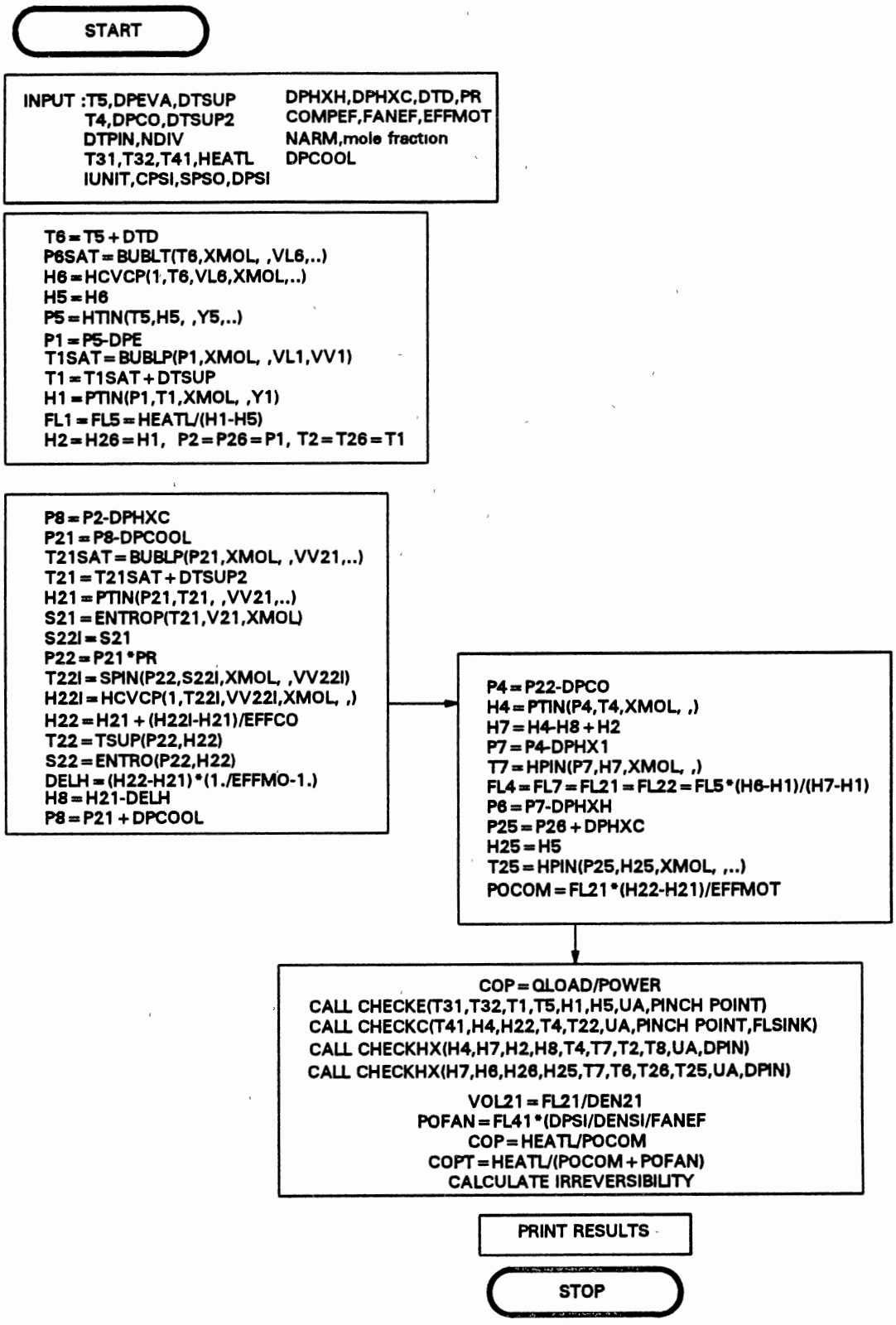


Figure 57. Flow Chart for Cycle 5

leaving the evaporator with a fixed mass flow rate. The heat discharged through the condenser, Q_{cond}, the sum of Q_{evap} , the compressor power W_{comp}, and the pump power W^{pump} (if a liquid pump is required) all depend on the working fluids and the cycles. The sink stream entering the condenser is assumed to be at ambient temperature. The heat sink flow rate can be varied to keep a minimum pinch point temperature in the condenser.

The following assumptions are applied for each cycle.

- A compressor isentropic efficiency of 0.75.

- A pump isentropic efficiency of 0.75 .

- The inlet and outlet temperatures of the source side are kept constant.

- A pinch point temperature of at least 5°F for every heat exchanger is assumed.

- The inlet temperature of the sink side is kept constant.

- An electric motor efficiency 0.85 is assumed. The motor is cooled and lubricated by the refrigerant. The bearing losses, seal losses, and motor losses in the compressor are accounted for in the overall system performance.

- All heat exchangers used in the model are of counter flow configuration to take advantage of the change in refrigerant temperature. To estimate the total conductance (UA) of the heat exchangers, the NARM's are assumed to be in equilibrium in every segment of the heat exchangers and the conductance calculations are based on to the temperature difference between the hot and the cold side of the heat exchangers only.

- A condenser fan efficiency of 0.75 is assumed.

- A 0.5 psi pressure drop is assumed for every heat exchanger.

- For air-cooled condenser systems, a 0.062 psi (2 inches of water) pressure drop for the sink side of the condenser is assumed.

CHAPTER V

DISCUSSION OF SIMULATION RESULTS

In previous chapters, simulation models have been developed to provide tools for evaluating refrigerant mixtures and identifying the acceptable NARM's and cycles which provide the best performance for vapor compression cycles. In this chapter, the models are used to estimate the performance of different working fluids for two operating conditions:

Case I : Heat sink temperature at 100°F, and the heat source temperature is changed from 70°F to 40°F (30°F temperature drop).

Case II : Heat sink temperature at 100°F, and the heat source temperature is changed from 55°F to 40°F (15°F temperature drop).

The results presented in this study are based on a one ton of cooling load (200 Btu/min). The system performances are presented in two forms:

- The coefficient of performance of vapor compression cycle (COP) alone. Only the electric power consumption for the compressor and the pump (if required by the cycle) are included.

$$COP = \frac{\text{Evaporator load}}{\text{Compressor+(Pump) Power consumption}} \qquad (5.1)$$

- The system coefficient of performance (COPT) which include the electric power consumption for the vapor compression cycle and the condenser fan.

$$COPT = \frac{\text{Evaporator load}}{\text{Compressor+Cond. fan+(Pump) Power consumption}} \quad (5.2)$$

The power consumption required to conduct the heat source stream through the evaporator is not included in the comparison because this value is the same for every cycle.

5.1 Methodology to Select the Optimum NARM's and Cycles

5.1.1 Parameters

As mentioned in Chapter IV, the parameters taken for the cycle analysis are:

- Cycle 1 :refrigerant, refrigerant inlet evaporator temperature, refrigerant outlet condenser temperature, refrigerant outlet evaporator degree of superheat, and outlet condenser degree of subcool.

- Cycle 2 :refrigerant, refrigerant inlet evaporator temperature, refrigerant outlet condenser temperature, refrigerant outlet evaporator degree of superheat, outlet condenser degree of subcool, first stage pressure ratio, and temperature drop across the expansion valve (which effects the size of subcooler).

- Cycle 3 : NARM, composition, NARM inlet evaporator temperature, NARM outlet condenser temperature, NARM outlet evaporator degree of superheat, and outlet condenser degree of subcool.

- Cycle 4 : NARM composition, NARM inlet evaporator temperature, NARM outlet condenser temperature, NARM outlet evaporator degree of superheat, first stage compressor pressure ratio, temperature drop across the expansion valve, and the temperature difference between

outlet of the hot side and inlet of the cold side of subcooler SUB 2 (which effects the size of subcooler SUB2).

- Cycle 5 : NARM composition, NARM inlet evaporator temperature, NARM outlet condenser temperature, NARM outlet evaporator degree of superheat, temperature drop across the expansion valve.

- Cycle 6 : NARM composition, NARM inlet evaporator temperature, NARM outlet condenser temperature, NARM outlet evaporator degree superheat, first stage pressure ratio, temperature drop across the expansion valve and the temperature difference between outlet of the hot side and inlet of the cold side of subcooler SUB 2.

The required parameters to establish the cycle operating point are summarized in Table V.

5.1.2 Search Technique for NARM and Cycle Selection

The objective of the simulation model is to search for the optimal combinations of refrigerant mixtures and the cycles. Therefore, the solution concentrates on the coefficient of performance. However other characteristics of the system are also recorded for the consideration of working fluid selection. These include compressor pressure ratio, compressor suction volume flow rate, total heat exchanger conductance, etc. Because of the complexity of the cycles and refrigerant property subroutines, a generic optimization method to maximize the coefficient of performance is not applied in this study. Instead the optimum solutions are obtained by an exhaustive search technique. Predetermined ranges of independent variables are defined. The simulation model calculates the system performance, pressure ratio, compressor volume flow rate and total heat exchanger conductance etc.for all combination of cycle operating condition in a "DO LOOP"

TABLE V

REQUIRED PARAMETERS FOR THE CYCLES

	Symbol	Cycle					
		1	2	3	4	5	6
Refrigerant 1	R1	X	X	X	X	X	X
Refrigerant 2	R2			X	X	X	X
NARM mol fraction of R1	XMOL			X	X	X	X
Inlet evaporator temperature	T5	X	X	X	X	X	X
Temp. drop across expa. valve	TDT		X		X	X	X
Temp. diff. at one side SUB 2	DTHX				X		X
Outlet evap. degree superheat	DTSUP	X	X	X	X	X	X
Outlet cond. degree subcool	T4	X	X	X			
First stage pressure ratio	PR1		X		X		

structure. The DO LOOP structure is shown in Figure 58 for a pure refrigerant R1 or a binary refrigerant mixture R1,R2.

5.1.3 Limitation Range of the Parameters

The limitations of operating conditions and the step sizes are as follows :

- The heat source outlet temperature, T31, considered in this application is 40°F, with a minimum 5°F pinch point temperature. Therefore, the maximum refrigerant evaporator inlet temperature is 35°F. The step size of the inlet evaporator temperature chosen for the cycle evaluation is 1°F ($T5 \leq 35°F$ and $\Delta T5 = 1°F$).

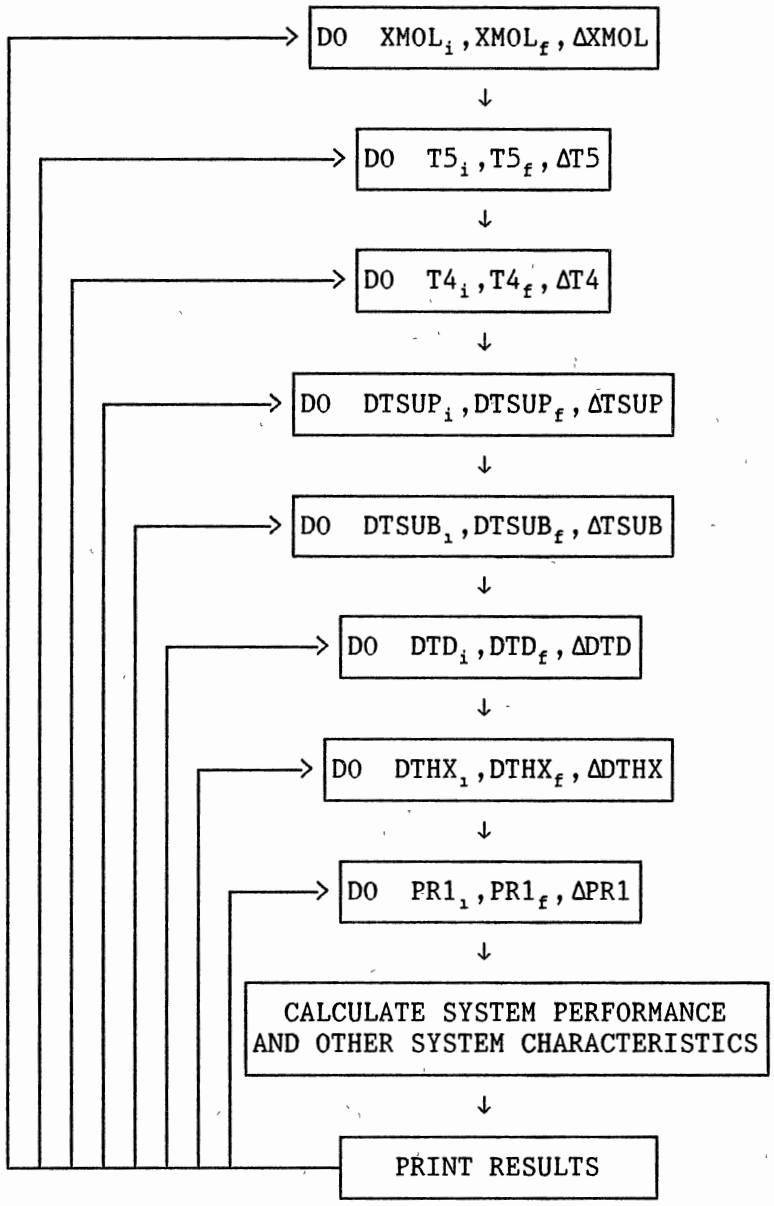


Figure 58. DO LOOP to Search for the Optimum NARM's

Parameters of the DO LOOP change from the initial value with subscript "i" to the final value with subscript "f" with a step size Δ.

- The heat sink inlet temperature, T41, considered in this application is 100°F, with a minimum 5°F pinch point temperature. Therefore, the minimum refrigerant condenser outlet temperature T4 is 105°F. The step size of the outlet condenser temperature chosen for the evaluation is 1°F ($T4 \geq 105°F$ and $\Delta T4 = 1°F$).

- All other values related to the temperature chosen for calculation are in step size of minimum 1°F.

- The minimum step size for the compressor pressure ratio, ΔPR, is 0.01.

- The minimum step size for mole fraction composition, ΔXMOL, is 0.01.

- All refrigerants in the NIST database, except refrigerants with a high ODP, are evaluated.

5.2 The Optimum NARM's and Cycles for Case I

For the cycles operated with pure refrigerants (Cycle 1 and Cycle 2), ten pure refrigerants from the NIST database were selected as the working fluids. The remaining 8 refrigerants were not selected because of their high ODP or their critical temperatures are too low for the applications in this study. Although, R-12 was banned by the Montreal Protocol, the performance of Cycle 1 and Cycle 2 with R-12 is shown for comparison purposes only.

For the cycles operated with NARM's, all promising mixtures with low ODP were selected as the working fluids. The mixtures are given in Figures 3.5, 3.6, 5.10 and 5.11.

5.2.1 Result of Simulation Models

The results of the simulation are summarized for the best combination of mixtures and the cycles. The criteria to select the optimum refrigerants are based on the COPT (including power consumption for the condenser fan). However, the other characteristics of the system are included, including the pressure ratio, total conductance of heat exchangers, evaporator pressure, compressor inlet volume flow rate etc.

Table VI shows the characteristics of Cycle 1 and pure refrigerants, for the maximum COPT.

Table VII shows the characteristics of Cycle 2 and pure refrigerants, for the maximum COPT.

Table VIII shows the characteristics of Cycle 3 and the optimum refrigerant mixtures, for the maximum COP and COPT.

Table IX shows the characteristics of Cycle 4 and the optimum refrigerant mixtures, for maximum COP and COPT.

Table X shows the characteristics of Cycle 5 and the optimum refrigerant mixtures, for the maximum COP and COPT.

Table XI shows the characteristics of Cycle 6 and the optimum refrigerant mixtures, for the maximum COP and COPT.

The data from from the above Tables (Table VI to Table XI) for the optimum system performance for an air cooled condenser system.

For a liquid cooled condenser system, the power consumption for the sink side pump is much less than the condenser fan power consumption of an air cooled condenser system. Therefore, the coefficient of vapor compression cycle (COP) alone (COP) better reflect the real power consumption for the liquid cooled system. For the NARM

vapor compression cycles, the maximum COP and COPT usually occur at the same operating conditions (refrigerant inlet evaporator temperature and refrigerant outlet condenser temperature). However, for pure refrigerants, the optimum COP and COPT occur at different operating conditions, usually at the same evaporating temperature but different condensing temperatures. The maximum COP occurs when the condensing temperature is the lowest (105°F).

Table XII shows the characteristics of Cycle 1 and pure refrigerants, for the maximum COP.

Table XIII shows the characteristics of Cycle 2 and pure refrigerants, for maximum COPT.

TABLE VI

CASE I: CHARACTERISTICS OF CYCLE 1 FOR HIGHEST COPT

REFRIGERANT	OUTLET COND. TEMP. (F)	COND. DEGREE SUBC.	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP. PRES. RATIO	COMP. 1 SUCTION FLOW RATE (CFM)	CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
R12	105.	15.0	35.	47.0	3.70	3.7	2.74	2.23	31.0	62.0
R22	105.	15.0	35.	77.0	3.63	2.3	2.71	2.22	26.0	55.0
R123	105.	16.0	35.	5.0	6.65	25.4	2.56	2.10	31.0	64.0
R124	105.	17.0	35.	25.0	4.58	6.3	2.44	2.02	31.0	64.0
R125	105.	16.0	35.	103.0	3.58	2.3	2.29	1.91	34.0	65.0
R134	105.	18.0	35.	35.0	4.58	4.4	2.67	2.21	30.0	59.0
R134a	105.	18.0	35.	45.0	4.36	3.6	2.63	2.19	30.0	57.0
R142b	105.	16.0	35.	35.0	4.51	6.7	2.78	2.25	31.0	64.0
R152a	105.	17.0	35.	41.0	4.27	3.7	2.75	2.26	28.0	58.0
RC270	105.	17.0	35.	53.0	3.74	3.1	2.77	2.29	27.0	57.0

TABLE VII

CASE I: CHARACTERISTICS OF CYCLE 2 FOR HIGHEST COPT

REFRIGERANT	OUTLET COND. TEMP. (F)	COND. DEGREE SUBC.	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP. PRES. RATIO	COMPRESSOR SUCTION FLOW RATE 1 (CFM) 2		CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
R12	105.	17.0	35.	47.0	3.80	3.0	2.4	2.87	2.34	32.0	59.0
R22	105.	15.0	35.	77.0	3.64	1.9	1.5	2.91	2.34	32.0	62.0
R123	105.	15.0	35.	5.0	6.53	22.1	11.0	2.71	2.16	33.0	70.0
R124	105.	17.0	35.	25.0	4.57	5.1	3.6	2.62	2.14	33.0	64.0
R125	105.	17.0	35.	103.0	3.62	2.0	0.9	2.37	1.97	35.0	64.0
R134	105.	18.0	35.	35.0	4.57	3.6	2.5	2.86	2.33	32.0	60.0
R134a	105.	18.0	35.	45.0	4.36	2.9	2.2	2.84	2.34	33.0	56.0
R142b	105.	17.0	35.	35.0	4.57	5.5	3.9	2.92	2.35	32.0	61.0
R152a	105.	17.0	35.	41.0	4.28	3.2	2.1	2.93	2.37	31.0	61.0
RC270	105.	18.0	35.	53.0	3.79	2.7	2.0	2.90	2.38	30.0	61.0

TABLE VIII

CASE I: CHARACTERISTICS OF CYCLE 3 FOR HIGHEST COPT

REFRIGERANT	OUTLET COND. TEMP. (F)	COND. DEGREE SUBC. (F)	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP. PRES. RATIO	COMP. 1 SUCTION FLOW RATE (CFM)	CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
.94 R22/.06 R123	105.	13.0	35.	71.0	3.57	2.3	2.84	2.52	35.0	33.0
.58 R124/.42 R123	105.	0.0	35.	15.0	4.10	9.2	3.01	2.47	55.0	54.0
.50 R124/.50 R125	105.	0.0	35.	24.0	3.36	3.2	2.83	2.32	63.0	57.0
.60 R134/.40 R123	105.	2.0	35.	20.7	4.13	6.7	3.23	2.84	71.0	31.0
.78 R134a/.22 R123	109.	3.0	35.	33.1	4.04	4.5	3.03	2.71	61.0	28.0
.55 R142b/.45 R123	105.	0.0	35.	13.4	4.11	10.2	3.32	2.64	57.0	58.0
.54 R142b/.46 R125	105.	0.0	35.	51.0	3.41	3.3	3.09	2.60	63.0	44.0
.73 R152a/.27 R123	105.	5.0	35.	29.1	3.88	4.9	3.24	2.87	61.0	29.0
.85 RC270/.15 R123	105.	11.0	35.	44.0	3.58	3.6	3.07	2.74	46.0	29.0

TABLE IX

CASE I: CHARACTERISTICS OF CYCLE 4 FOR HIGHEST COPT

REFRIGERANT	OUTLET COND. TEMP. (F)	COND. DEGREE SUBC.	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP. PRES. RATIO	COMPRESSOR SUCTION FLOW RATE (CFM) 1	2	CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
.94 R22/.06 R123	116.	0.0	35.	72.0	3.44	1.9	1.6	3.04	2.64	44.0	37.0
.59 R124/.41 R123	105.	0.0	35.	17.0	3.77	6.7	5.1	3.37	2.80	70.0	45.0
.67 R124/.33 R125	105.	0.0	35.	49.0	3.08	2.5	2.8	3.24	2.56	66.0	61.0
.70 R134/.30 R123	108.	0.0	35.	26.0	3.72	4.5	3.4	3.60	3.07	66.0	25.0
.82 R134a/.18 R123	112.	0.0	35.	37.0	3.74	3.3	2.3	3.42	2.98	60.0	34.0
.67 R142b/.33 R123	105.	0.0	35.	17.0	3.68	6.8	5.6	3.75	2.78	58.0	69.0
.58 R142b/.42 R125	105.	0.0	35.	55.0	2.99	2.4	2.0	3.65	2.99	77.0	45.0
.79 R152a/.21 R123	111.	0.0	35.	33.0	3.68	3.3	2.3	3.51	3.02	59.0	34.0
.87 RC270/.13 R123	113.	0.0	35.	47.0	3.33	1.6	2.1	3.37	2.90	52.0	36.0

TABLE X

CASE I: CHARACTERISTICS OF CYCLE 5 FOR HIGHEST COPT

REFRIGERANT	OUTLET COND. TEMP. (F)	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP PRES. RATIO	COMP. SUCTION FLOW RATE (CFM)	OUTLET EVAP. QUALITY	CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
.60 R22/.40 R123	105.	35.	45.0	2.05	6.0	0.54	3.31	2.91	108.0	30.0
.50 R23/.50 R134	105.	35.	186.0	1.64	2.1	0.49	3.31	2.71	117.0	50.0
.53 R123/.47 R134	105.	35.	19.4	3.75	7.6	0.82	3.30	2.90	83.0	31.0
.50 R123/.50 RC270	105.	35.	29.0	2.32	8.6	0.56	3.14	2.70	90.0	39.0
.72 R125/.28 R123	105.	35.	72.0	2.35	3.6	0.63	2.93	2.70	93.0	21.0
.53 R134a/.47 R123	105.	35.	25.0	3.16	7.2	0.70	3.19	2.82	83.0	31.0
.55 R152a/.45 R123	105.	35.	25.0	3.31	6.8	0.75	3.29	2.88	75.0	31.0

TABLE XI

CASE I: CHARACTERISTICS OF CYCLE 6 FOR HIGHEST COPT

REFRIGERANT	OUTLET COND. TEMP. (F)	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP PRES. RATIO	COMP. SUCTION FLOW RATE (CFM)	OUTLET EVAP. QUALITY	CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
.50 R22/.50 R123	105.	35.	38.0	1.93	7.5	0.46	3.49	2.91	119.0	40.0
.50 R123/.50 R134	105.	35.	20.0	3.71	6.8	0.84	3.56	3.00	78.0	39.0
.50 R123/.50 R134a	105.	35.	24.0	3.55	6.3	0.67	3.38	2.84	81.0	42.0
.50 R123/.50 R152a	105.	35.	23.0	3.53	6.4	0.71	3.48	2.94	82.0	39.0
.59 R123/.41 RC270	105.	35.	25.0	2.06	11.0	0.50	3.31	2.64	102.0	57.0
.70 R125/.30 R123	105.	35.	71.0	2.10	4.0	0.60	3.00	2.71	97.0	27.0
.60 R134/.40 R23	105.	35.	153.0	1.64	2.5	0.44	3.40	2.62	129.0	65.0

TABLE XII

CASE I: CHARACTERISTICS OF CYCLE 1 FOR HIGHEST COP

REFRIGERANT	OUTLET COND. TEMP. (F)	COND. DEGREE SUBC.	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP. PRES. RATIO	COMP. 1 SUCTION FLOW RATE (CFM)	CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
R12	105.	1.0	35.	47.0	3.07	3.7	3.23	0.75	48.0	753.0
R22	105.	1.0	35.	77.0	3.02	2.3	3.21	0.82	36.0	667.0
R123	105.	1.0	35.	5.0	5.13	25.4	3.07	1.04	51.0	472.0
R124	105.	1.0	35.	25.0	3.61	6.3	2.96	0.68	54.0	844.0
R125	105.	1.0	35.	103.0	2.95	2.3	2.70	0.65	54.0	862.0
R134	105.	1.0	35.	35.0	3.57	4.4	3.24	0.71	52.0	788.0
R134a	105.	1.0	35.	45.0	3.42	3.6	3.18	0.71	50.0	816.0
R142b	105.	1.0	35.	35.0	3.62	6.7	3.30	0.83	50.0	663.0
R152a	105.	1.0	35.	41.0	3.40	3.7	3.31	0.72	43.0	808.0
RC270	105.	1.0	35.	53.0	3.03	3.1	3.36	0.77	53.0	737.0

TABLE XIII

CASE I: CHARACTERISTICS OF CYCLE 2 FOR HIGHEST COP

REFRIGERANT	OUTLET COND. TEMP. (F)	COND. DEGREE SUBC. (F)	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP. PRES. RATIO	COMPRESSOR SUCTION FLOW RATE 1 (CFM) 2		CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
R12	105.	1.0	35.	47.0	3.07	3.0	2.4	3.48	0.75	55.0	772.0
R22	105.	1.0	35.	77.0	3.02	1.9	1.5	3.48	0.81	49.0	704.0
R123	105.	1.0	35.	5.0	5.13	22.1	11.0	3.20	1.03	56.0	486.0
R124	105.	1.0	35.	25.0	3.62	5.1	3.6	3.21	0.68	58.0	851.0
R125	105.	1.0	35.	103.0	2.95	2.0	0.9	2.84	0.69	59.0	807.0
R134	105.	1.0	35.	35.0	3.60	3.6	2.5	3.52	0.73	57.0	801.0
R134a	105.	1.0	35.	45.0	3.42	2.9	2.2	3.48	0.71	57.0	833.0
R142b	105.	1.0	35.	35.0	3.62	5.5	3.9	3.53	0.83	56.0	677.0
R152a	105.	1.0	35.	41.0	3.40	3.2	2.1	3.58	0.70	54.0	842.0
RC270	105.	1.0	35.	53.0	3.03	2.7	2.0	2.58	0.75	51.0	774.0

5.2.2 Discussion of Simulation Results

The system performance and characteristics for different cycle arrangements, refrigerant constituents, compositions and cycle states have been established. To select optimum NARMs and cycles, the following criteria were considered:

- The system operates with safe refrigerants.

- The system coefficient of performance (COPT) should be high. These criteria are most important. At higher coefficient of performance, the system requires less of electric power and therefore the operating cost as well as the system demand Kw is reduced.

- The total compression pressure ratio should be low. In this study, the compressor efficiency was assumed 0.75. However, in reality, at lower pressure ratio, the compressor operates more efficiently, and therefore the total power consumption is reduced. Furthermore at lower pressure ratio, the compressor can be designed as a single stage unit operating at lower speed. This improves compressor reliability and reduces complexity.

- The total heat exchanger conductance (UA) should be low. This reduces the heat exchanger size and makes it compact. In addition there is reduction in heat exchanger cost as well as the system weight.

- The volume flow rate at compressor inlet should be low. Line pressure drops were not included in this study. With smaller vapor volume flowrate, the pressure drop in the suction line, and the compressor inlet port is low. As a result, the system compresion pressure ratio is low therefore a smaller compressor size is satisfactory.

- The evaporator pressure should preferably be higher than atmospheric pressure to avoid leakage of air into the system and permit easier detection of refrigerant leaks.

- The condensing pressure is should be low. This allows the hardware construction with light weight materials and reduces the tendency of refrigerant leakage.

Figure 59 illustrates the COPT of the pure refrigerants as the working fluid for cycle 1 and cycle 2 and the highest COPT of Non-azeotropic refrigerant mixtures associated with the NARM cycles. The system coefficient of performance, COPT, reflects the actual electric power consumption for the vapor compression system operated with air as the heat sink because the power consumption for the condenser fan is included in the total power consumption. The pressure drop of the heat sink through the condenser is assumed to be 0.06 psid (2.0 inches of water).

Figure 60 shows the related COP (coefficient of vapor compression alone) for the same mixtures, cycles, and operating condition as in Figure 59.

Figure 61 shows the related compressor pressure ratio for the same mixtures, cycles, and operating condition as in Figure 59.

Figure 62 shows the related total heat exchanger conductances for the same mixtures, cycles, and operating condition as in Figure 59.

Figure 63 shows the volume flow rate at the first-stage compressor for the same mixtures, cycles, and operating condition as in Figure 59.

Figure 64 shows the operating pressure range (from the lowest pressure to highest pressure) for the same mixtures, cycles, and operating condition as in Figure 59.

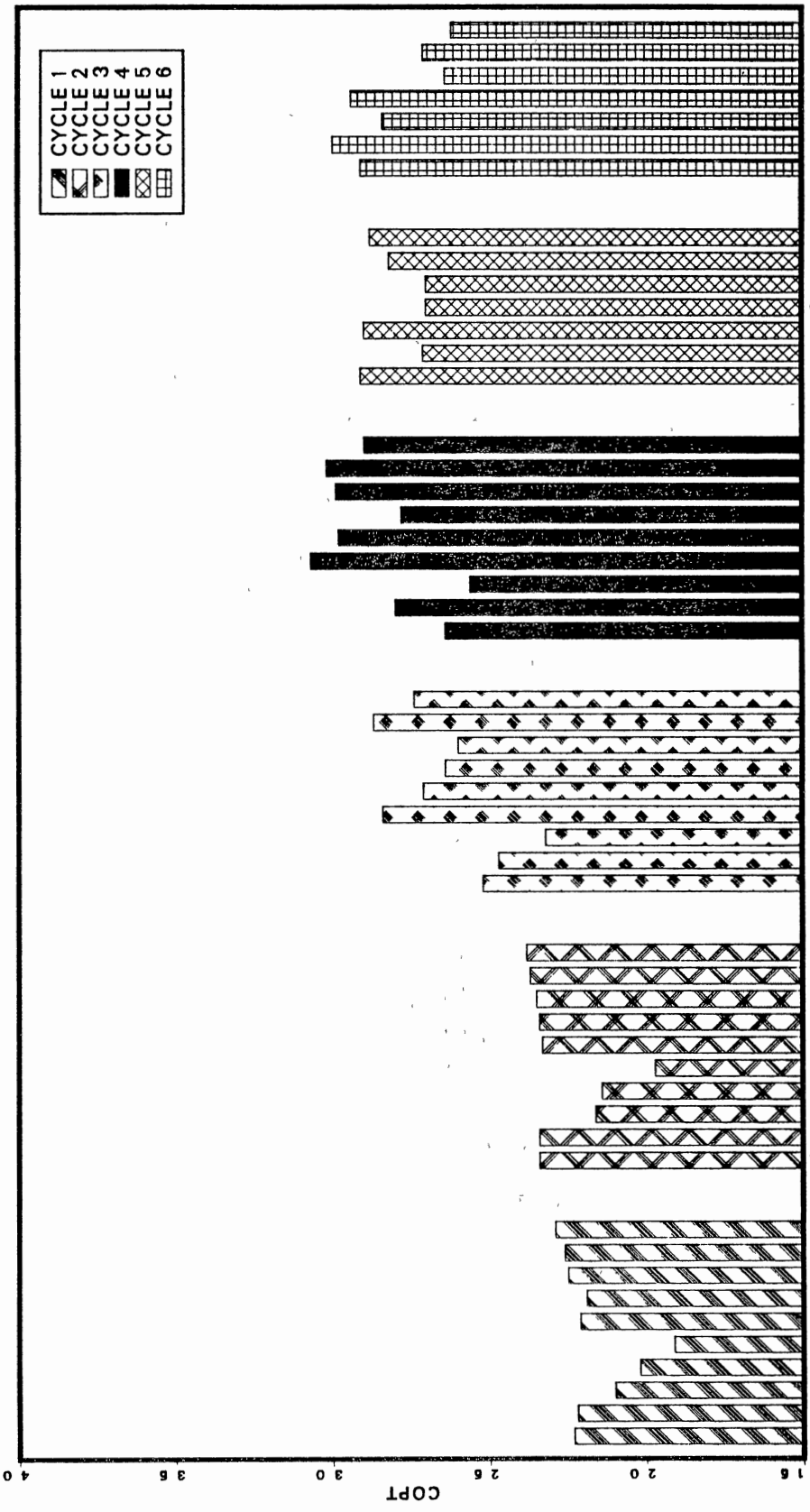


Figure 59. 30°F Source Temperature Loss- System Coefficient of Performance (COPT)

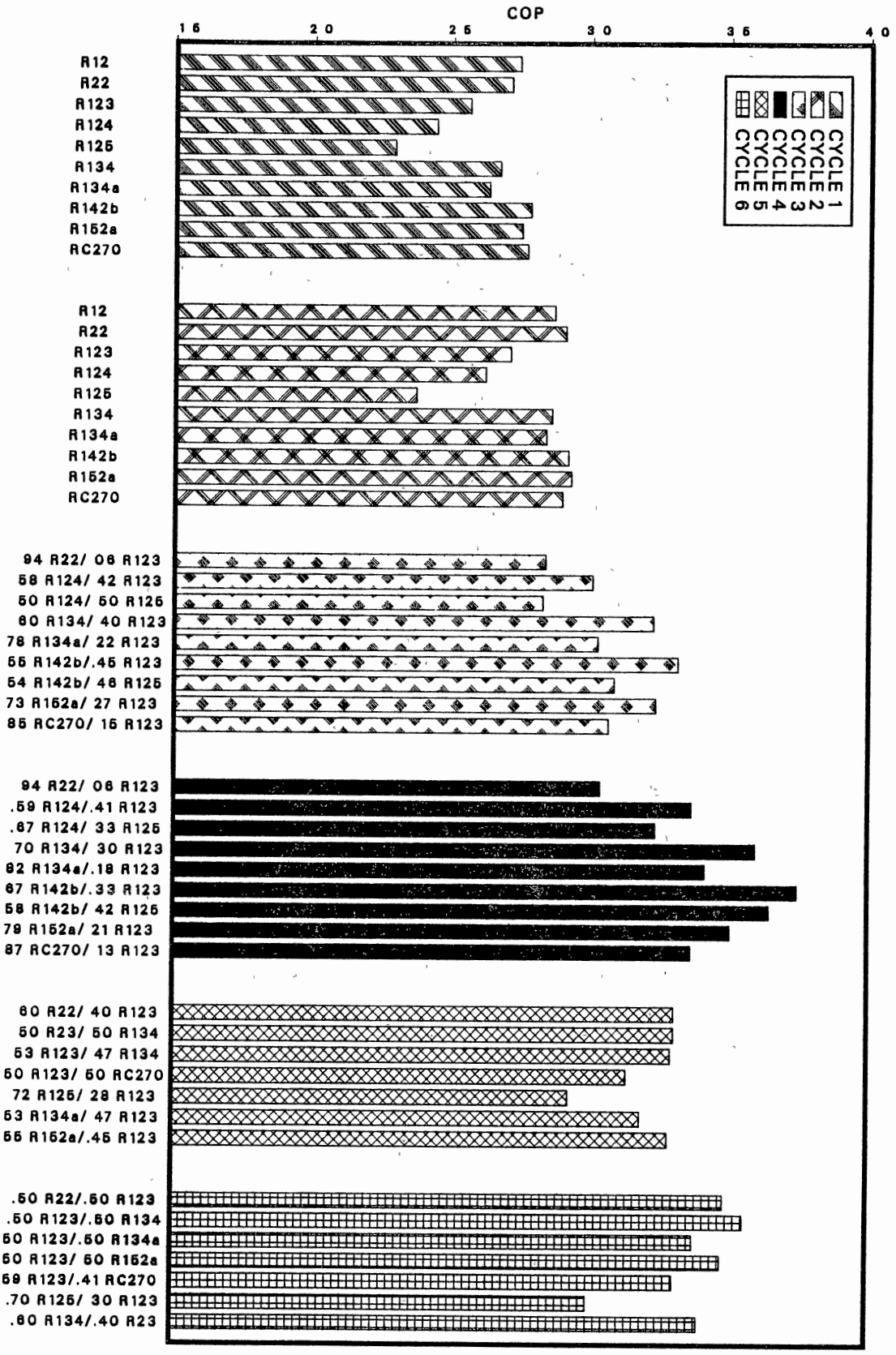


Figure 60. 30°F Source Temperature Loss- VCS Coefficient of Performance (COP)

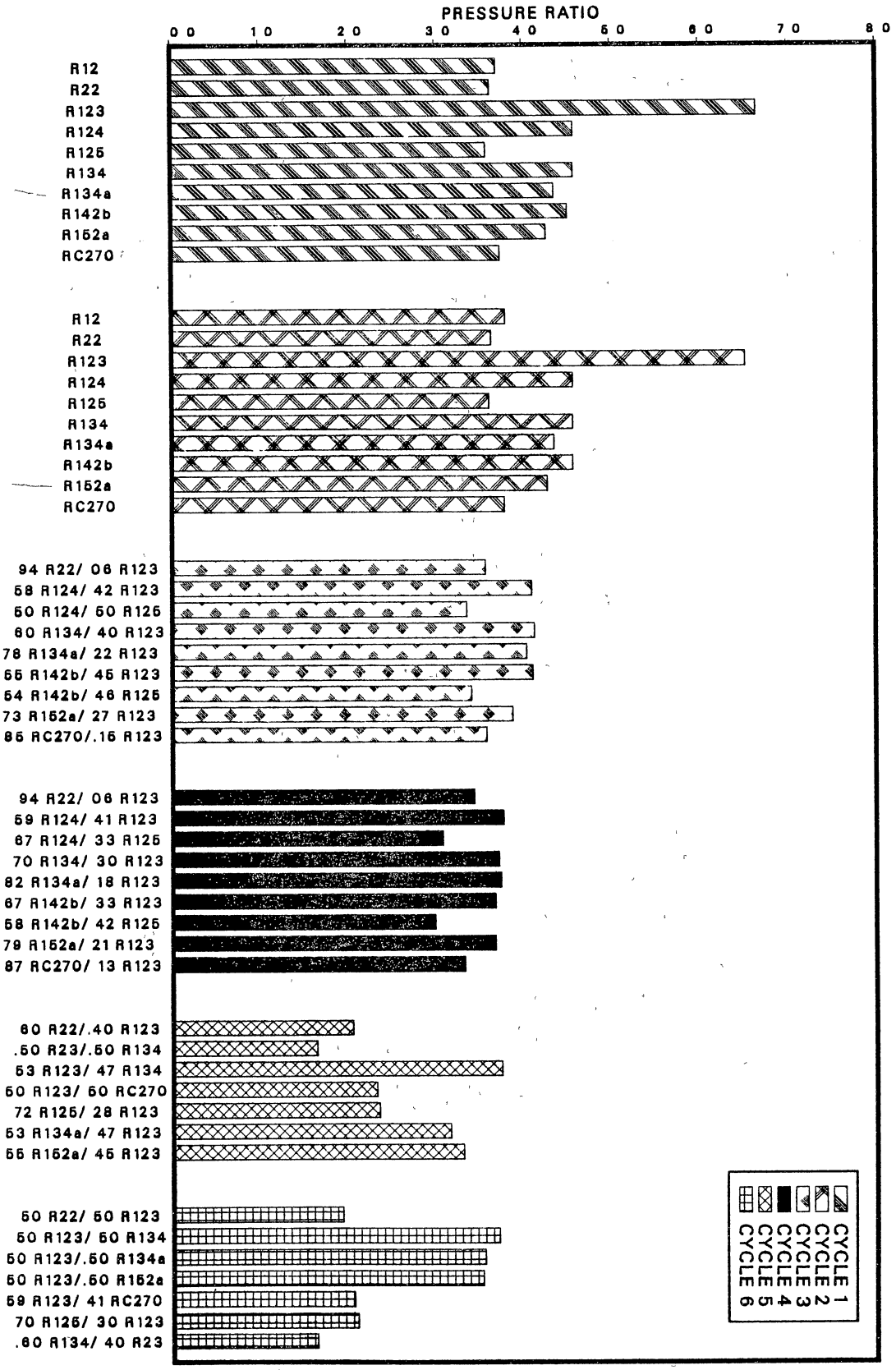


Figure 61. 30°F Source Temperature Loss- Compressor Pressure Ratio

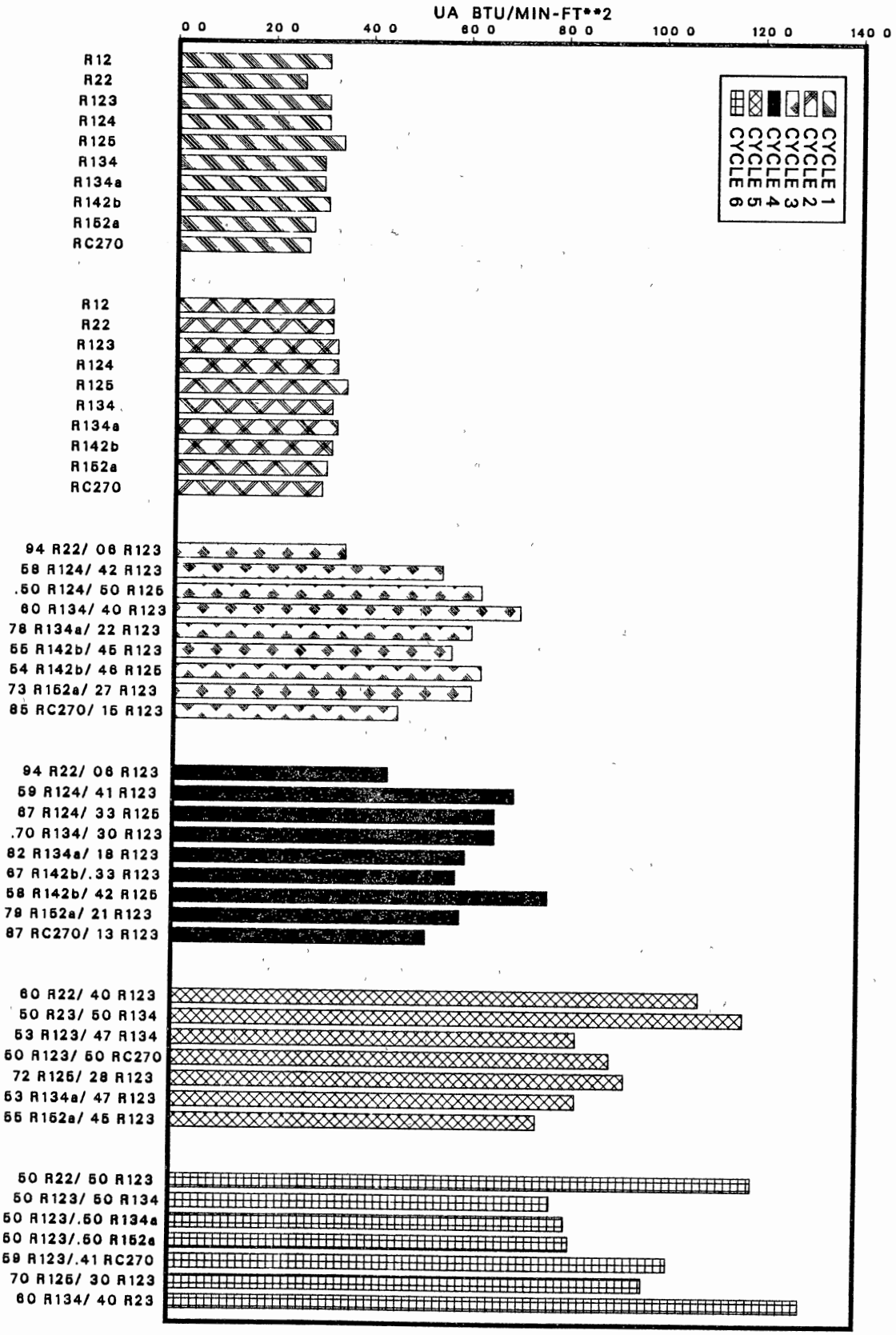


Figure 62. 30°F Source Temperature Loss- Total Heat Exchanger Conductance

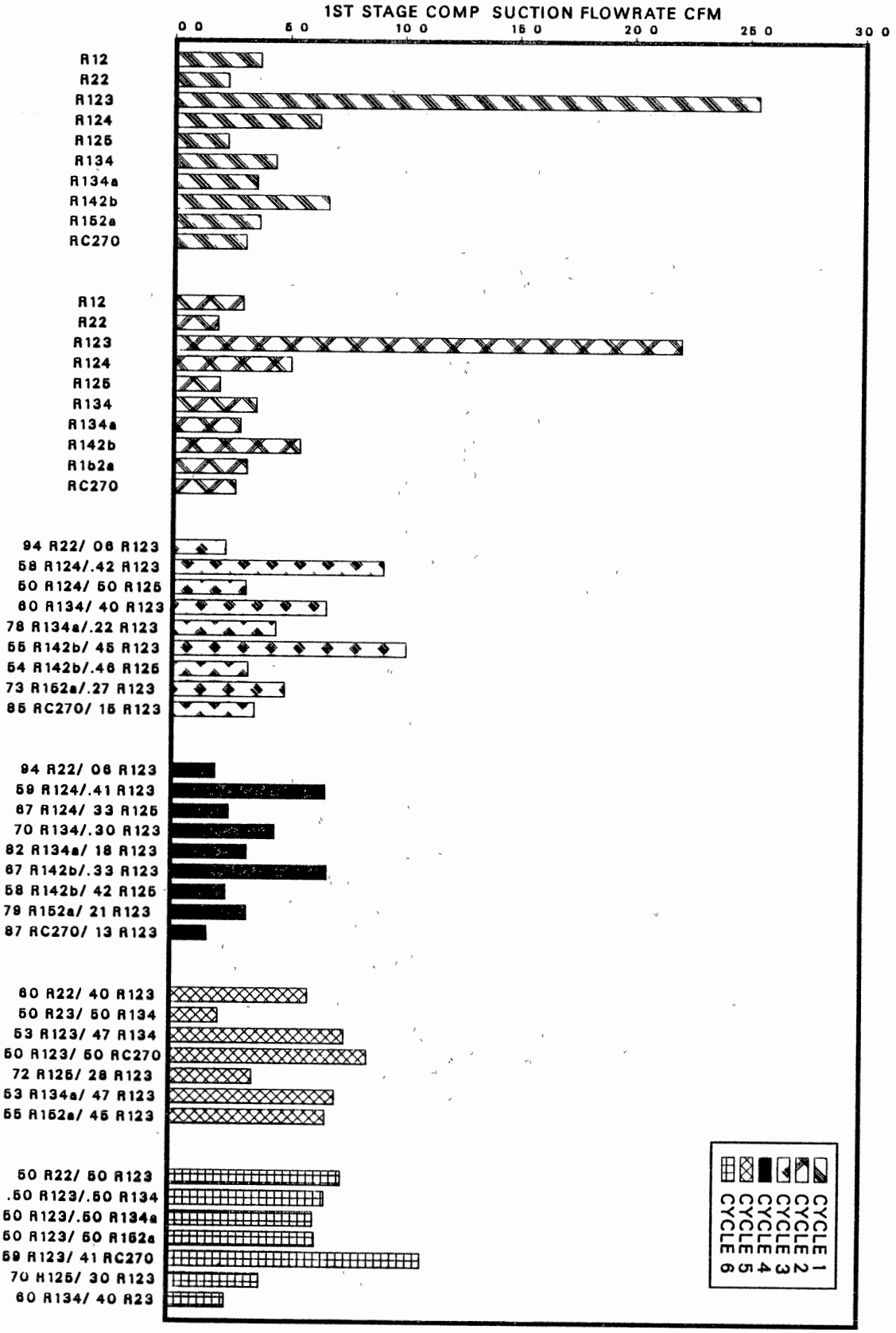


Figure 63. 30°F Source Temperature Loss- First Stage Compressor Volume Flow Rate

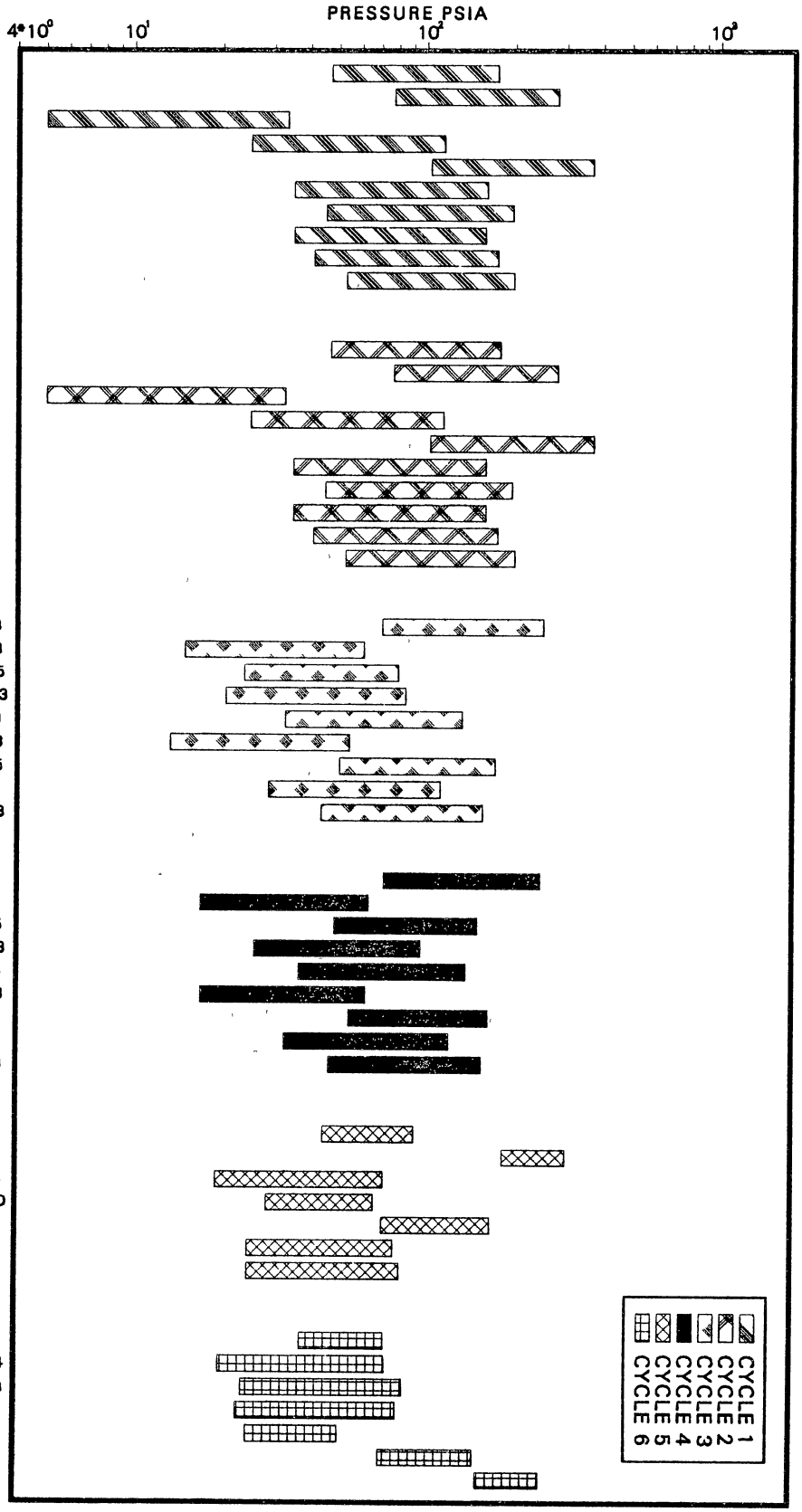


Figure 64. 30°F Source Temperature Loss- Operating Pressure Range

To compare the system performance, the cycles are divided into three groups: a group for pure refrigerants, a group for NARM's with single phase at the evaporator outlet, and a group for NARM's with two-phase at the evaporator outlet.

The group of pure refrigerant includes Cycle 1 and Cycle 2. For Cycle 1, COPT varies between 1.91 for R-125 to 2.29 for RC-270. For Cycle 2, COPT varies between 1.97 for R-125 to 2.38 for RC-270. In general using the pure refrigerants as the working fluid for two-stage compressor, the COPT is increased approximately 5.0% compared with the COPT of the same working fluid for a single-stage compressor systems. The performance improvement is summarized in Table XIV.

The group for NARM's with single phase vapor at the evaporator outlet includes Cycle 3 and Cycle 4. For Cycle 3, COPT varies from 2.32 for mixture 0.5/0.5 R-124/R-125 to 2.87 for mixture 0.73/0.27 R-152a/R-123. For Cycle 4, COPT varies from 2.56 for mixture 0.67/0.33 R-124/R-125, to 3.02 for mixture 0.79/0.21 R-152a/R-123. The COPT of NARM's

TABLE XIV

COPT IMPROVEMENT, CYCLE 2 COMPARED TO CYCLE 1

Refrigerant	COPT		Percent Improvement
	Cycle 1	Cycle 2	
R-12	2.23	2.34	4.9
R-22	2.22	2.34	5.4
R-123	2.10	2.16	2.9
R-124	2.02	2.14	5.9
R-125	1.91	1.97	3.1
R-134	2.21	2.33	5.4
R-134a	2.19	2.34	6.8
R-142b	2.25	2.35	4.4
R-152a	2.26	2.37	4.9
RC-270	2.29	2.38	3.9

TABLE XV

COPT IMPROVEMENT, CYCLE 4 COMPARED TO CYCLE 3

Refrigerant Mixtures	COPT		Percent Improvement
	Cycle 3	Cycle 4	
R-22/R-123	2.52	2.64	4.8
R-124/R-123	2.47	2.80	13.4
R-124/R-125	2.32	2.56	10.3
R-134/R-123	2.84	3.07	8.1
R-134a/R-123	2.71	2.98	10.0
R-142b/R-123	2.64	2.78	5.3
R-142b/R-125	2.60	2.99	15.0
R-152a/R-123	2.87	3.02	5.2
RC-270/R-123	2.74	2.90	5.8

operating with 2 stage compression is increased approximately 9.0% compared to the conventional system. The performance improvement is summarized in Table XV.

The group for NARM's with two-phase flow at the evaporator outlet includes Cycle 5 and Cycle 6. The variation of COPT for the NARM's operating with Cycle 5 were from 2.70 for 0.72/0.28 R-125/R-123 mixture, to 2.91 for 0.60/0.40 R-22/R-123 mixture. The variation of COPT for NARM's operating with Cycle 6 from 2.62 for 0.60/0.40 R-134/R-23 mixture, to 3.00 for 0.50/0.50 R-123/R-134. The improvements of Cycle 6 over Cycle 5 are not clear and depend on the refrigerant mixtures as shown in Table XVI.

The comparison between the system performance and pressure ratio for NARM cycles and pure refrigerant cycles can be made now. The comparison is based on three values of COPT, the COPT for NARM cycles and the other two COPT values for the pure refrigerant cycles in which

TABLE XVI

COPT IMPROVEMENT, CYCLE 6 COMPARED TO CYCLE 5

Refrigerant Mixtures	COPT		Percent Improvement
	Cycle 5	Cycle 6	
R-22/R123	2.91	2.91	0.0
R-23/R134	2.71	2.62	-3.3
R-123/R134	2.90	3.00	3.4
R-123/RC270	2.70	2.64	-2.2
R-125/R123	2.70	2.71	0.4
R-134a/R123	2.82	2.84	0.7
R-152a/R123	2.88	2.94	2.1

the pure refrigerants are the constituents of the NARM's. Then the percent improvement of NARM cycle over each pure refrigerant cycle can be calculated. Finally, the overall improvement of COPT of NARM cycle is the lower value.

For example, the COPT for Cycle 3 operating with mixture of 0.78 R-134a/0.22 R-123 was 2.71. The COPT for pure R-134a with Cycle 1 was 2.19, and the improvement is 23.7%. The COPT for pure R-123 with Cycle 1 is 2.10, with an improvement of 29.0%. Therefore, the overall improvement of R-134a/R-123 with Cycle 3 over pure refrigerant cycle is approximately 23.7%.

The average improvement of COPT for the optimum NARM's with Cycle 3 is approximately 20% higher than for pure refrigerant system. Note that not all NARM systems have better performance than pure systems.

A similar comparison of the compressor pressure ratio shows that the pressure ratio for the best NARM's with Cycle 3 is approximately 7% less than those for pure refrigerant systems.

TABLE XVII

COPT IMPROVEMENT, CYCLE 3 COMPARED TO CYCLE 1

	COPT			Percent Improvement	
	Mixture	Pure Refrig. Ref. 1	Ref. 2	Ref.1	Ref.2
.94 R22/.06 R123	2.52	2.22	2.10	13.5	20.0
.58 R124/.42 R123	2.47	2.02	2.10	22.3	17.6
.50 R124/.50 R125	2.32	2.02	1.91	14.9	21.5
.60 R134/.40 R123	2.84	2.21	2.10	28.5	35.2
.78 R134a/.22 R123	2.71	2.19	2.10	23.7	29.0
.55 R142b/.45 R123	2.64	2.25	2.10	17.3	25.
.54 R142b/.46 R125	2.60	2.25	1.91	15.6	36.1
.73 R152a/.27 R123	2.87	2.26	2.10	27.0	36.7
.85 RC270/.15 R123	2.74	2.29	2.10	19.7	30.5

The COPT's, and percent improvement of the best NARM's for Cycle 3 in this study over pure refrigerant systems are given in Table XVII.

The compressor pressure ratio and percent pressure ratio reduction of the optimum NARM's for Cycle 3 in this study over pure refrigerant systems are given in Table XVIII.

As the comparison between the optimum NARM's of Cycle 5 and pure refrigerants of Cycle 1 shows, the performance, COPT, of the optimum NARM operating with Cycle 5 in this study is approximately 28% higher than the pure refrigerant system with Cycle 1.

A comparison of the pressure ratio shows that the pressure ratio for the optimum NARM's with Cycle 5 are approximately 30% less than the pressure ratio for a pure refrigerant system with Cycle 1.

The COPT's and improvement of the optimum NARM's for Cycle 5 in this study compared to pure refrigerant systems are given in Table IXX.

TABLE XVIII

PRESSURE RATIO REDUCTION, CYCLE 3 COMPARED TO CYCLE 1

	Pressure Ratio			Percent Reduction	
	Mixture	Pure Refrig. Ref. 1	Ref. 2	Ref.1	Ref.2
.94 R22/.06 R123	3.57	3.63	6.65	-1.7	-46.3
.58 R124/.42 R123	4.10	4.58	6.65	-10.5	-38.3
.50 R124/.50 R125	3.36	4.58	3.58	-26.6	-6.1
.60 R134/.40 R123	4.13	4.58	6.65	-9.8	-37.9
.78 R134a/.22 R123	4.04	4.36	6.65	-7.3	-39.2
.55 R142b/.45 R123	4.11	4.51	6.65	-8.9	-38.2
.54 R142b/.46 R125	3.41	4.51	3.58	-24.4	-4.7
.73 R152a/.27 R123	3.88	4.27	6.65	-9.1	-41.7
.85 RC270/.15 R123	3.58	3.74	6.65	-4.3	-46.2

TABLE IXX

COPT IMPROVEMENT, CYCLE 5 COMPARED TO CYCLE 1

	COPT			Percent Improvement	
	Mixture	Pure Refrig. Ref. 1	Ref. 2	Ref.1	Ref.2
.60 R22/.40 R123	2.91	2.22	2.10	31.1	38.6
.53 R123/.47 R134	2.90	2.10	2.21	38.1	31.2
.50 R123/.50 RC270	2.70	2.10	2.29	28.6	17.9
.72 R125/.28 R123	2.70	1.91	2.10	41.4	28.6
.53 R134a/.47 R123	2.82	2.19	2.10	28.8	34.3
.55 R152a/.45 R123	2.88	2.26	2.10	27.4	37.1

The compressor pressure ratio and pressure ratio reduction of the optimum NARM's for Cycle 5 in this study compared to pure refrigerant systems are given in Table XX.

TABLE XX

PRESSURE RATIO REDUCTION, CYCLE 5 COMPARED TO CYCLE 1

	Pressure Ratio			Percent Reduction	
		Pure Refrig.			
	Mixture	Ref. 1	Ref. 2	Ref.1	Ref.2
.60 R22/.40 R123	2.05	3.63	6.65	-43.5	-69.2
.53 R123/.47 R134	3.75	6.65	4.58	-43.6	-18.1
.50 R123/.50 RC270	2.32	6.65	3.74	-65.1	-38.0
.72 R125/.28 R123	2.35	3.58	6.65	-34.4	-64.7
.53 R134a/.47 R123	3.16	4.36	6.65	-27.5	-52.5
.55 R152a/.45 R123	3.31	4.27	6.65	-22.5	-50.2

A comparison between Cycle 6 and Cycle 1 shows that the performance, COPT, of the optimum NARM's operating with Cycle 6 in this study is 28% higher than the pure refrigerant system. Note that not all NARM systems have better performance than pure system.

A comparison of the pressure ratio shows that the compressor pressure ratios for the optimum NARM's are approximately 31% less than for those of pure refrigerant systems.

The COPT's, and improvement of the optimum NARM's for Cycle 6 in this study over pure refrigerant systems are given in Table XXI.

Similarly, the compressor pressure ratio and pressure ratio reduction of the optimum NARM's for Cycle 6 compared to pure refrigerant systems are given in Table XXII.

Cycle 2 for a pure refrigerant is now compare with Cycle 4 for NARM. The reason to compare these cycles is that these cycles need a two-stage compressor for operating.

TABLE XXI

COPT IMPROVEMENT, CYCLE 1 COMPARED TO CYCLE 1

	COPT			Percent Improvement	
	Mixture	Pure Refrig. Ref. 1	Ref. 2	Ref.1	Ref.2
.50 R22/.50 R123	2.91	2.22	2.10	31.1	38.6
.50 R123/.50 R134	3.00	2.10	2.21	42.9	35.7
.59 R123/.41 RC270	2.64	2.10	2.29	25.7	15.3
.70 R125/.30 R123	2.71	1.91	2.10	41.9	29.0
.50 R123/.50 R134a	2.84	2.10	2.19	35.2	29.7
.50 R123/.50 R152a	2.94	2.10	2.26	40.0	30.1

TABLE XXII

PRESSURE RATIO REDUCTION OF CYCLE 6 COMPARED TO CYCLE 1

	Pressure Ratio			Percent Reduction	
	Mixture	Pure Refrig. Ref. 1	Ref. 2	Ref.1	Ref.2
.50 R22/.50 R123	1.93	3.63	6.65	-46.8	-71.0
.50 R123/.50 R134	3.71	6.65	4.58	-44.2	-19.0
.59 R123/.41 RC270	2.06	6.65	3.74	-69.0	-44.9
.70 R125/.30 R123	2.10	3.58	6.65	-41.3	-68.4
.50 R123/.50 R134a	3.55	6.65	4.36	-46.6	-18.6
.50 R123/.50 R152a	3.53	6.65	4.27	-46.9	-17.3

A comparison between Cycle 4 and Cycle 2 shows that the performance, COPT, of the optimum NARM operating with cycle 4 is approximately 24% higher than a pure refrigerant system. A similar comparison of the pressure ratio shows that the pressure ratio for the

COPT IMPROVEMENT, CYCLE 4 COMPARED TO CYCLE 2

	COPT			Percent Improvement	
		Pure Refrig.			
	Mixture	Ref. 1	Ref. 2	Ref.1	Ref.2
.94 R22/.06 R123	2.64	2.34	2.16	12.8	22.2
.59 R124/.41 R123	2.80	2.14	2.16	30.8	29.6
.67 R124/.33 R125	2.56	2.14	1.97	19.6	29.9
.70 R134/.30 R123	3.07	2.33	2.16	31.8	42.1
.82 R134a/.18 R123	2.98	2.34	2.16	27.4	38.0
.67 R142b/.33 R123	2.78	2.35	2.16	18.3	28.7
.58 R142b/.42 R125	2.99	2.35	1.97	27.2	51.8
.79 R152a/.21 R123	3.02	2.37	2.16	27.4	39.8
.87 RC270/.13 R123	2.90	2.38	2.16	21.8	34.3

optimum NARM's are approximately 15% less than the pure refrigerant system.

The COPT's and improvement of the optimum NARM's for Cycle 4 compared to pure refrigerants with Cycle 2 are given in Table XXIII.

The compressor pressure ratio and percent pressure ratio reduction of the optimum NARM's for Cycle 4 compared to pure refrigerant system with Cycle 2 are given in Table XXIV.

A comparison between the optimum NARM's with Cycle 4 and pure refrigerants with Cycle 1 shows that the performance, COPT, of the optimum NARM operating with Cycle 4 is 30% higher than the pure refrigerant system with Cycle 1.

A comparison of the pressure ratio shows that the compressor pressure ratios for the optimum NARM's with Cycle 4 are approximately 14% less than the pure refrigerant systems with Cycle 1.

The COPT's and improvement of the optimum NARM's for Cycle 4 compared to pure refrigerant systems are given in Table XXV.

TABLE XXIV

PRESSURE RATIO REDUCTION, CYCLE 4 COMPARED TO CYCLE 2

	PRESSURE RATIO			Percent reduction	
		Pure Refrig.			
	Mixture	Ref. 1	Ref. 2	Ref.1	Ref.2
.94 R22/.06 R123	3.44	3.64	6.53	-5.5	-47.3
.59 R124/.41 R123	3.77	4.57	6.53	-17.5	-42.3
.67 R124/.33 R125	3.08	4.57	3.62	-32.6	-14.9
.70 R134/.30 R123	3.72	4.57	6.53	-18.6	-43.0
.82 R134a/.18 R123	3.74	4.36	6.53	-14.2	-42.7
.67 R142b/.33 R123	3.68	4.57	6.53	-19.5	-43.6
.58 R142b/.42 R125	2.99	4.57	3.62	-34.6	-17.4
.79 R152a/.21 R123	3.68	4.28	6.53	-14.0	-43.6
.87 RC270/.13 R123	3.33	3.79	6.53	-12.1	-49.0

TABLE XXV

COPT IMPROVEMET, CYCLE 4 COMPARED TO CYCLE 1

Cycle 4 Mixtures	COPT			Percent Improvement	
		Pure Refrig.			
	Mixture	Ref. 1	Ref. 2	Ref.1	Ref.2
.94 R22/.06 R123	2.64	2.22	2.10	18.9	25.7
.59 R124/.41 R123	2.80	2.02	2.10	38.6	33.3
.67 R124/.33 R125	2.56	2.02	1.91	26.7	34.0
.70 R134/.30 R123	3.07	2.21	2.10	38.9	46.2
.82 R134a/.18 R123	2.98	2.19	2.10	36.1	41.9
.67 R142b/.33 R123	2.78	2.25	2.10	23.6	32.4
.58 R142b/.42 R125	2.99	2.25	1.91	32.9	56.5
.79 R152a/.21 R123	3.02	2.10	2.26	43.8	33.6
.87 RC270/.13 R123	2.90	2.29	2.10	26.6	38.1

The compressor pressure ratio and pressure ratio reduction of the optimum NARM's for Cycle 4 compared to pure refrigerant systems are given in Table XXVI.

Finally, the comparison of total conductance of heat exchangers for the six cycles is given in Table XXVII

TABLE XXVI

PRESSURE RATIO REDUCTION, CYCLE 4 COMPARED TO CYCLE 1

Cycle 4 Mixtures	Pressure ratio			Percent Reduction	
	Mixture	Pure Refrig. Ref. 1	Ref. 2	Ref.1	Ref.2
.94 R22/.06 R123	3.44	3.63	6.65	-5.2	-48.3
.59 R124/.41 R123	3.77	4.58	6.65	-17.7	-43.3
.67 R124/.33 R125	3.08	4.58	3.58	-32.8	-14.0
.70 R134/.30 R123	3.72	4.58	6.65	-18.8	-44.1
.82 R134a/.18 R123	3.74	4.36	6.65	-14.2	-43.8
.67 R142b/.33 R123	3.68	4.51	6.65	-18.4	-44.7
.58 R142b/.42 R125	2.99	4.51	3.58	-33.7	-16.5
.79 R152a/.21 R123	3.68	6.65	4.27	-44.7	-13.8
.87 RC270/.13 R123	3.33	3.74	6.65	-11.0	-49.9

TABLE XXVII

INCREMENT OF HEAT EXCHANGER CONDUCTANCE COMPARED TO CYCLE 1

	Number of Heat Exchanger			Average Total UA Btu/min-F	Percent Increase vs Cycle1
	Evaporator	Condenser	Subcooler		
Cycle 1	1	1	0	30.	0.0
Cycle 2	1	1	1	32	7.0
Cycle 3	1	1	0	57	90.0
Cycle 4	1	1	2	61	103.0
Cycle 5	1	1	2	92	206.0
Cycle 6	1	1	2	98	227.0

5.2.3 Results of Irreversibility Analysis

The performance of NARM vapor compression cycle be benefited by reducing the irreversibility in the heat exchangers. The temperature difference between the refrigerant and the second fluid in the heat exchangers needs to be as small as possible. In this study, the selection of optimum cycles and NARMs was based on the coefficient of performance with 5°F minimum pinch point temperatures in heat exchangers. Ideally the temperature difference between the refrigerant and the second fluid throughout heat exchanger should be 5°F. However, the temperature profiles are not parallel since the enthalpy-temperature characteristic of NARM are nonlinear. As a result a pinch point of 5°F for the optimum case may not always be achievable.

The procedure used to determine the irreversibility of individual components and cycle are described in Section 4.1. Typical working fluids for each cycle are selected to present the irreversibility analysis are summarized in Table XXVIII and Figures 65, 66, 67, 68, 69 and 70.

In general, the irreversibility of the individual NARM components was lower than the irreversibility of the individual pure refrigerant components. However, there are more components in a NARM cycle than in a pure refrigerant cycle. The results show that the irreversibility of the NARM evaporator is about 30% to 45% of the irreversibility of the pure refrigerant evaporator. The irreversibility of the NARM condenser is about 48% to 60% of the irreversibility of the pure refrigerant condenser. For a single-stage compression cycle, the irreversibility of the NARM compressor is about 75% to 84% of the irreversibility of the pure refrigerant compressor. The total irreversibility of the NARM

TABLE XXVIII

IRREVERSIBILITY ANALYSIS

	CYCLE 1 R134a		CYCLE 2 R134a		CYCLE 3 0.78/0.22 R134a/R123		CYCLE 4 0.82/0.18 R134a/R123		CYCLE 5 0.60/0.40 R22/R123		CYCLE 6 0.50/0.50 R22/R123	
	IRR	PER	IRR	PER	IRR	PER	IRR	PER	IRR	PER	IRR	PER
Evaporator	8.7	21.5	8.6	23.8	3.2	11.2	3.9	16.3	3.0	11.7	2.6	10.7
Condenser	6.2	15.4	5.7	15.9	3.4	12.0	3.1	13.1	3.2	12.6	3.0	12.1
Compressor	17.3	42.7	16.8	46.7	14.6	50.4	13.6	56.7	13.5	53.4	13.0	52.9
Expansion device	6.9	17.1	2.7	7.4	6.3	21.9	0.8	3.4	0.6	2.5	0.5	2.2
Motor cooling	1.4	3.3	0.5	1.5	1.3	4.5	0.4	1.5	1.1	4.2	0.2	0.6
Subcooler I			1.6	4.4			1.2	5.2	3.3	12.9	4.1	16.7
Subcooler II							0.7	3.1	0.7	2.8	0.8	3.3
Liquid pump											0.0	0.1
Flow mixing			0.1	0.3			0.2	0.7			0.3	1.4
Total	40.4	100.0	36.0	100.0	28.9	100.0	23.9	100.0	25.2	100.0	24.5	100.0

* IRR : irreversibility Btu/min. PER : percent for each component

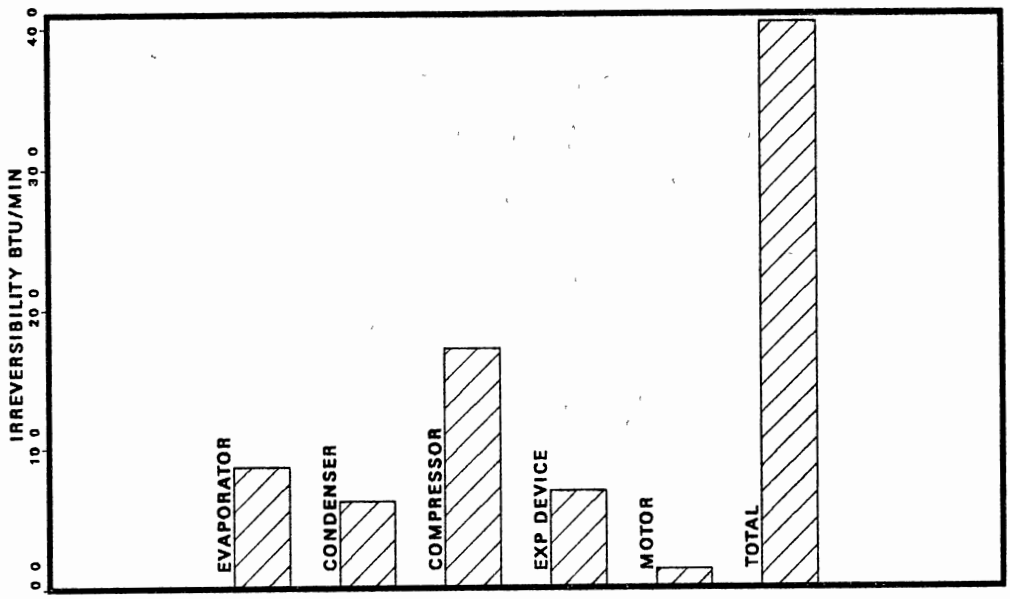


Figure 65. Irreversibility of Cycle 1 with R134a

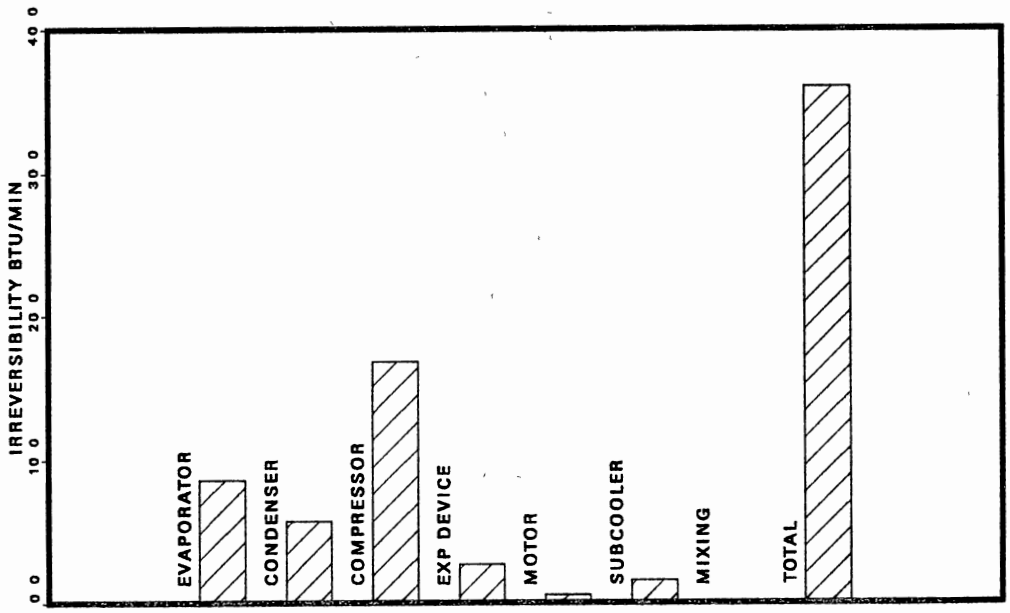


Figure 66. Irreversibility of Cycle 2 with R134a

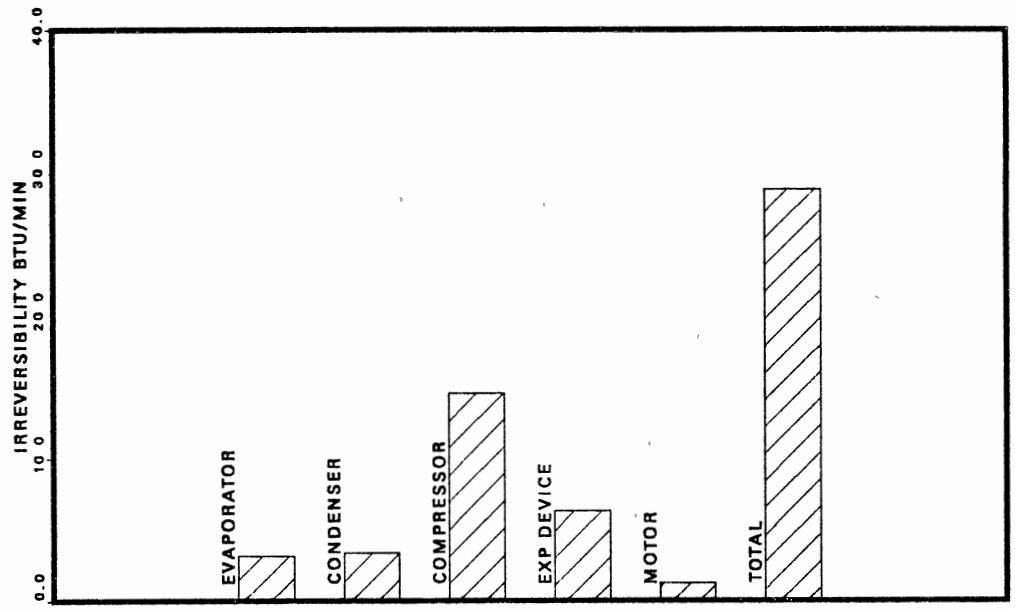


Figure 67. Irreversibility of Cycle 3 with .78 R134a/.22 R123

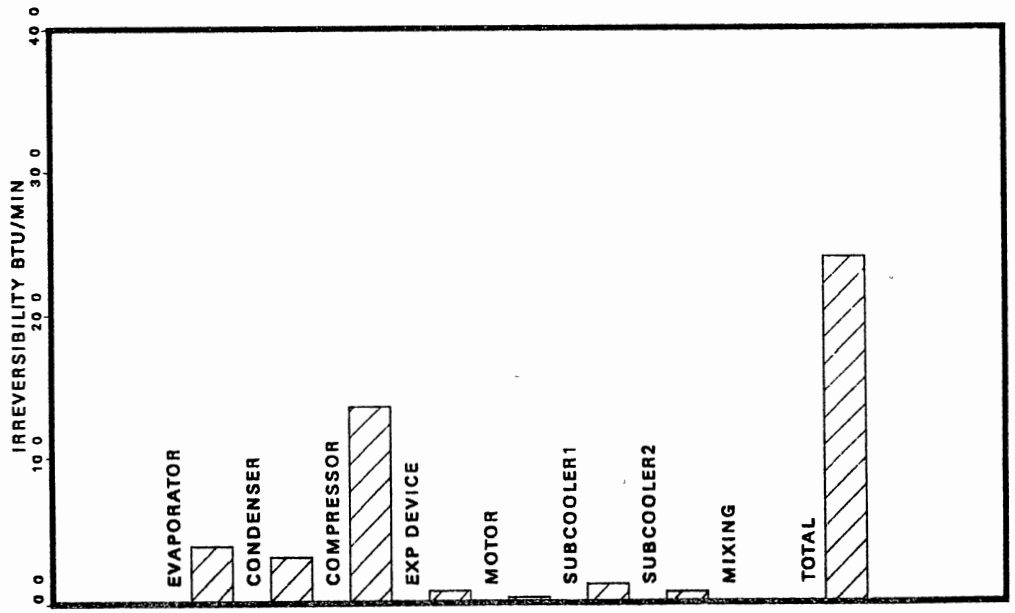


Figure 68. Irreversibility of Cycle 4 with .82 R134a/.28 R123

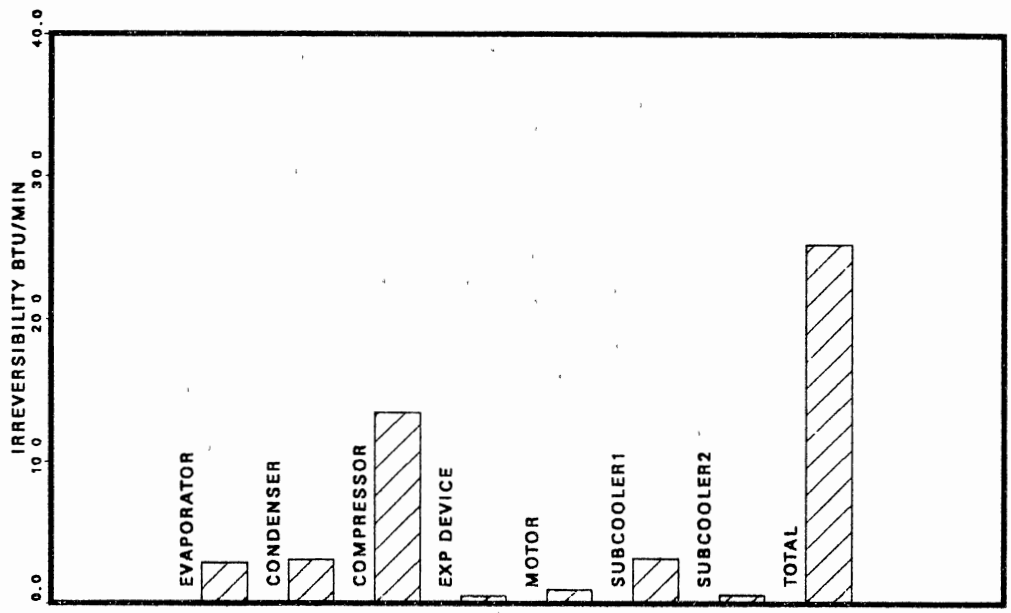


Figure 69. Irreversibility of Cycle 5 with .60R22/.40 R123

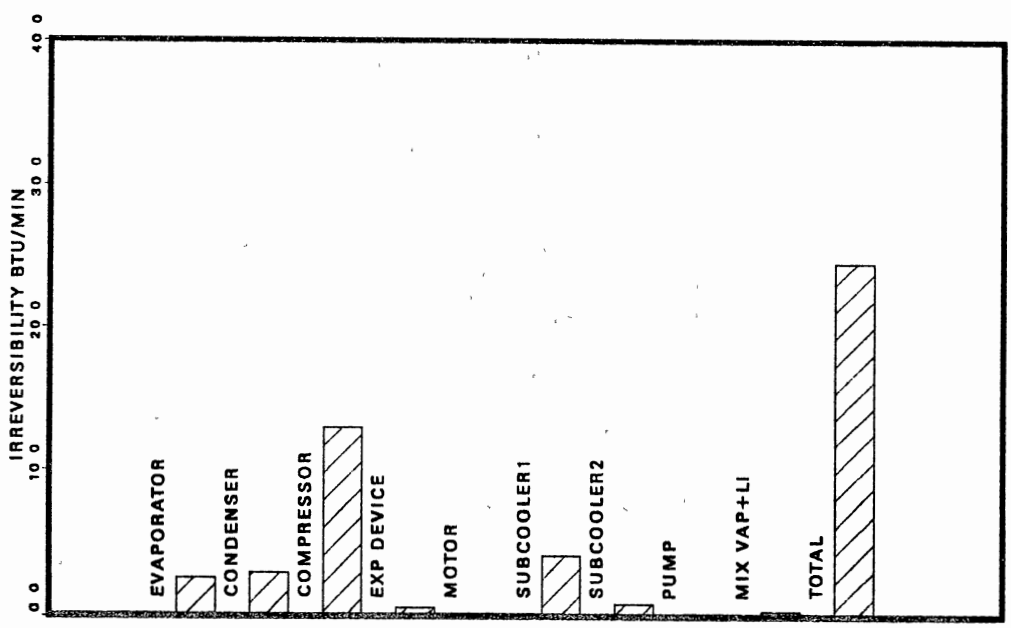


Figure 70. Irreversibility of Cycle 6 with .50 R22/.50 R123

expansion devices is about 7% to 9% of irreversibility of the pure refrigerant expansion devices. For a multi-stage compression cycle, the irreversibility of the NARM compressor is about 80% of the irreversibility of the pure refrigerant compressor. Finally, the irreversibility of the additional subcooler for the advance NARM cycles (Cycle 4, 5 and 6) is about 8% to 20% of total irreversibility.

5.3 The Optimum NARM's and Cycles for Case II

Case II, the heat sink temperature is 100°F, the heat source inlet temperature is 55°F, with 15°F temperature drop. For a pure refrigerant system (Cycle 1 and Cycle 2), the performance for 15°F heat source temperature drop are almost the same as the performance for 30°F temperature drop because the evaporating and condensing temperature are the same for two cases. Only the conductance of evaporators for the two cases are different. Therefore the performance for pure refrigerant systems are not included in this section.

5.3.1 Result of Simulation Models

The results of the simulation models are summarized for the optimum combination of mixtures and the cycles only.

Table XXIX shows the characteristics of Cycle 3 and the optimum refrigerants mixtures with the maximum COPT.

Table XXX shows the characteristics of Cycle 4 and the optimum refrigerants mixtures with the maximum COPT.

Table XXXI shows the characteristics of Cycle 5 and the optimum refrigerants mixtures with the maximum COPT.

Table XXXII shows the characteristics of Cycle 6 and the optimum refrigerants mixtures with the maximum COPT.

TABLE XXIX

CASE II: CHARACTERISTICS OF CYCLE 3 FOR HIGHEST COPT

REFRIGERANT	OUTLET COND. TEMP. (F)	COND. DEGREE SUBC. (F)	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP. PRES. RATIO	COMP. 1 SUCTION FLOW RATE (CFM)	CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
.50 R22/.50 R124	105.	0.0	35.	48.0	3.28	3.5	3.24	2.38	68.0	83.0
.50 R22/.50 R142b	105.	0.0	35.	46.0	3.30	3.5	3.29	2.56	72.0	64.0
.78 R124/.22 R123	105.	0.0	35.	55.0	3.38	3.2	2.99	2.38	86.0	57.0
.50 R124/.50 R125	105.	0.0	35.	24.0	3.36	3.2	2.83	2.32	63.0	57.0
.76 R125/.24 R142b	106.	0.0	35.	77.0	3.15	2.5	2.93	2.31	80.0	68.0
.88 R134/.12 R123	105.	4.0	35.	30.0	3.17	4.9	3.17	2.50	63.0	63.0
.58 R134/.42 R125	105.	0.0	35.	59.0	3.30	2.9	3.10	2.20	69.0	97.0
.93 R134a/.07 R123	105.	5.0	35.	41.0	3.70	3.7	3.10	2.46	59.0	63.0
.70 R142b/.30 R123	105.	0.0	35.	16.0	3.88	8.6	3.37	2.52	73.0	74.0
.91 R152a/.09 R123	105.	5.0	35.	37.0	3.67	4.0	3.19	2.55	55.0	59.0

TABLE XXX

CASE II: CHARACTERISTICS OF CYCLE 4 FOR HIGHEST COPT

REFRIGERANT	OUTLET COND. TEMP. (F)	COND. DEGREE SUBC.	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP. PRES. RATIO	COMPRESSOR SUCTION FLOW RATE 1 (CFM) 2		CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
.66 R22/.34 R142b	105.	0.0	35.	58.0	3.02	2.4	1.7	3.73	2.61	76.0	85.0
.50 R22/.50 R124	105.	0.0	35.	51.0	3.08	2.7	2.0	3.73	2.61	84.0	85.0
.82 R124/.18 R123	106.	0.0	35.	22.0	3.66	5.4	4.5	3.37	2.44	72.0	84.0
.78 R125/.22 R124	105.	0.0	35.	84.0	2.90	1.7	1.3	3.54	2.31	84.0	112.0
.83 R125/.17 R142b	106.	0.0	35.	88.0	2.93	1.7	1.5	3.45	2.43	78.0	89.0
.89 R134/.11 R123	108.	0.0	35.	31.0	3.69	3.9	2.9	3.49	2.59	66.0	74.0
.59 R134/.41 R125	105.	0.0	35.	63.0	3.06	2.1	1.7	3.32	2.30	83.0	99.0
.77 R142b/.23 R123	106.	0.0	35.	18.0	3.69	6.3	4.8	3.67	2.66	74.0	77.0

TABLE XXXI

CASE II: CHARACTERISTICS OF CYCLE 5 FOR HIGHEST COPT

REFRIGERANT	OUTLET COND. TEMP. (F)	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP PRES. RATIO	COMP. SUCTION FLOW RATE (CFM)	OUTLET EVAP. QUALITY	CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
.62 R22/.38 R123	105.	35.	46.0	1.74	8.8	0.37	2.89	2.44	139.0	48.0
.90 R134a/.10 R123	108.	35.	41.0	3.76	3.9	0.89	3.05	2.55	62.0	48.0
.80 R125/.20 R123	107.	35.	81.0	2.52	3.6	0.55	2.40	2.20	85.0	27.0
.90 R152a/.10 R123	109.	35.	37.0	3.67	4.2	0.95	3.15	2.53	58.0	58.0
.90 RC270/.10 R123	109.	35.	37.0	3.40	4.2	0.87	3.14	2.68	58.0	58.0

TABLE XXXII

CASE I I: CHARACTERISTICS OF CYCLE 6 FOR HIGHEST COPT

REFRIGERANT	OUTLET COND. TEMP. (F)	INLET EVAP. TEMP. (F)	INLET EVAP. PRESSURE (PSIA)	COMP PRES. RATIO	COMP. SUCTION FLOW RATE (CFM)	OUTLET EVAP. QUALITY	CYCLE COP	CYCLE COP INCLUDE FAN	TOTAL UA BTU/MIN-F	AIR SINK FLOW RATE LB/MIN
.60 R22/.40 R123	105.	35.	44.0	1.65	9.7	0.36	2.96	2.42	156.0	50.0
.90 R134a/.50 R123	108.	35.	41.0	3.56	4.0	0.86	3.11	2.38	70.0	74.0

5.3.2 Discussion of Simulation Results

Figure 71 illustrates the COPT of the pure refrigerant working fluids for Cycle 1 and Cycle 2 as well as and the highest COPT of the NARM's associated with the NARM cycles.

Figure 72 shows the COP (coefficient of vapor compression alone) for the same mixtures, cycles, and operating conditions as in Figure 71.

Figure 73 shows the related compressor pressure ratio for the same mixtures, cycles and operating condition as in Figure 71.

Figure 74 shows the related total heat exchanger conductances for the same mixtures, cycles and operating conditions as in Figure 71.

Figure 75 shows the volume flow rate at the first-stage of compressor for the same mixtures, cycles, and operating conditions as in Figure 71.

Figure 76 shows the operating pressure range (from the lowest pressure to highest pressure) for the same mixtures, cycles, and operating conditions as in Figure 71.

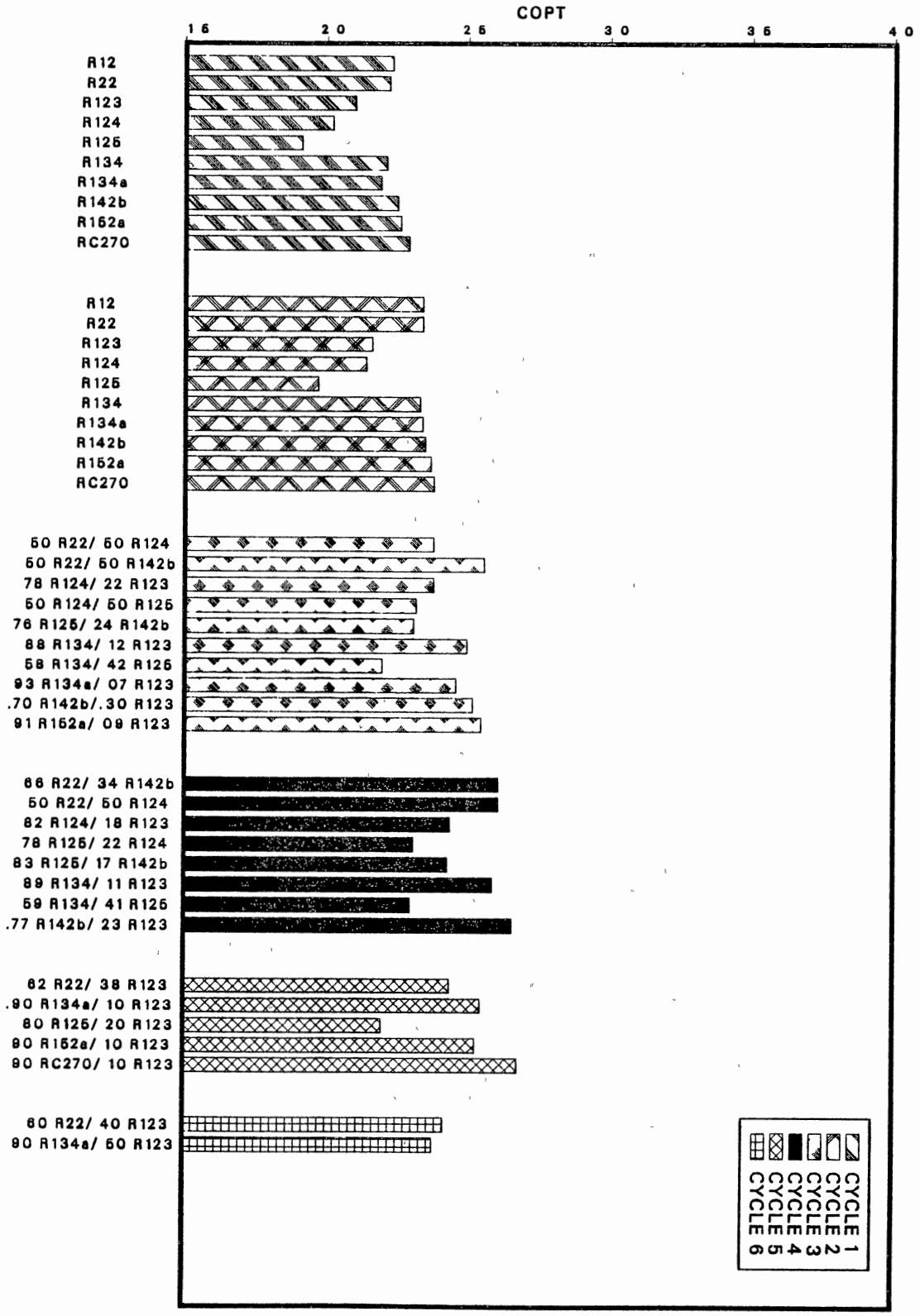


Figure 71. 15°F Source Temperature Loss- System Coefficient of Performance (COPT)

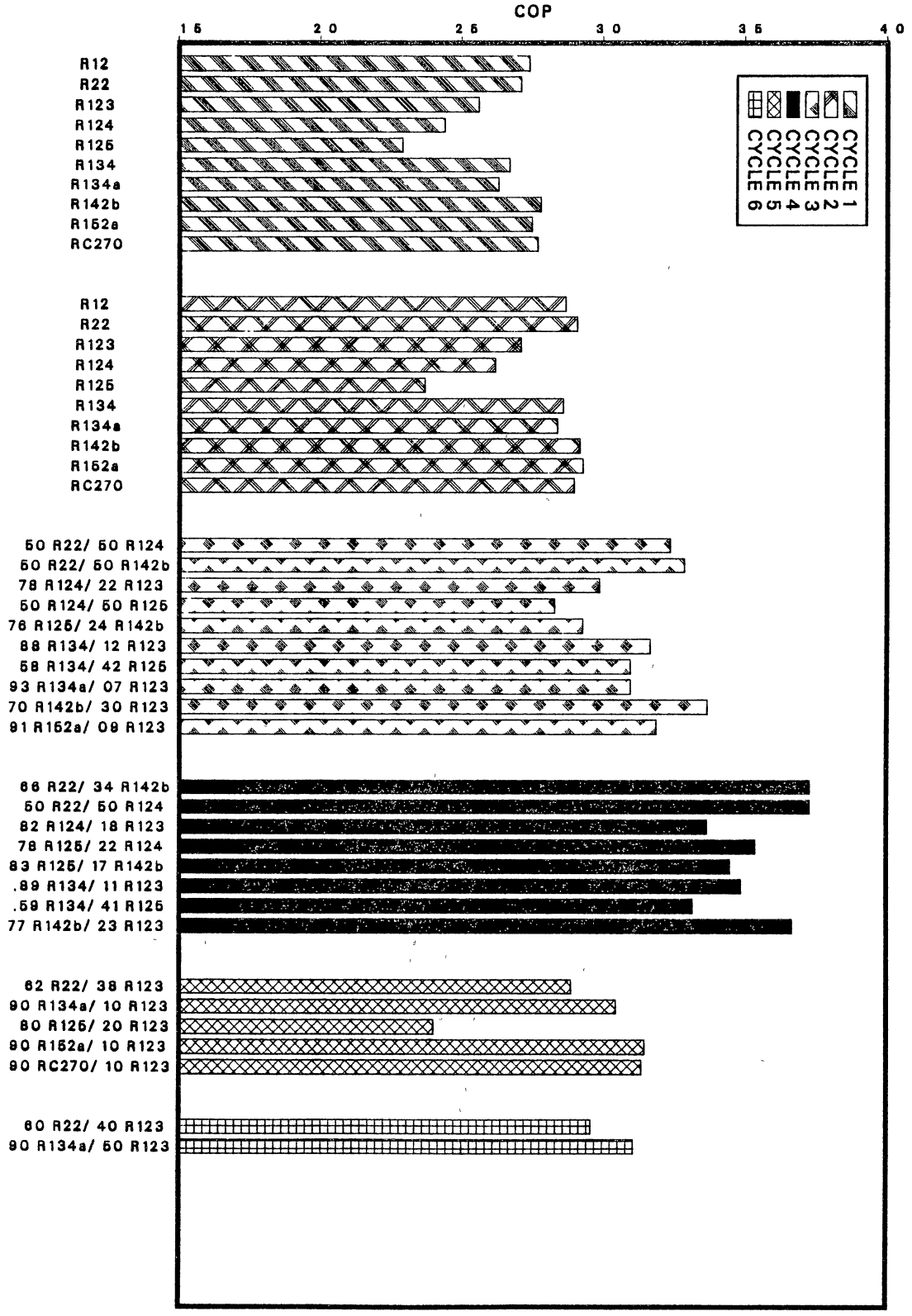


Figure 72. 15°F Source Temperature Loss- VCS Coefficient of Performance (COP)

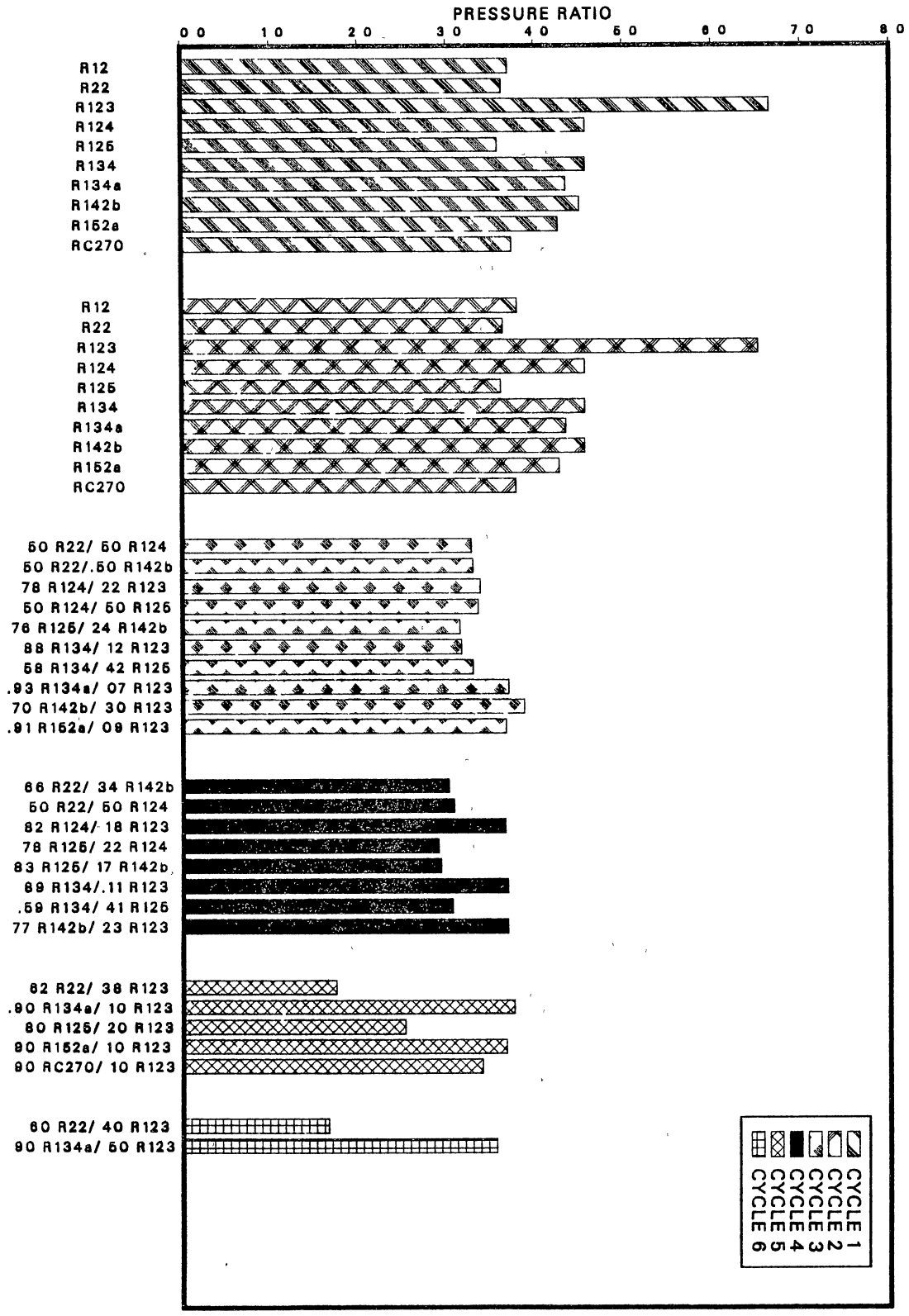


Figure 73. 15°F Source Temperature Loss- Compressor Pressure Ratio

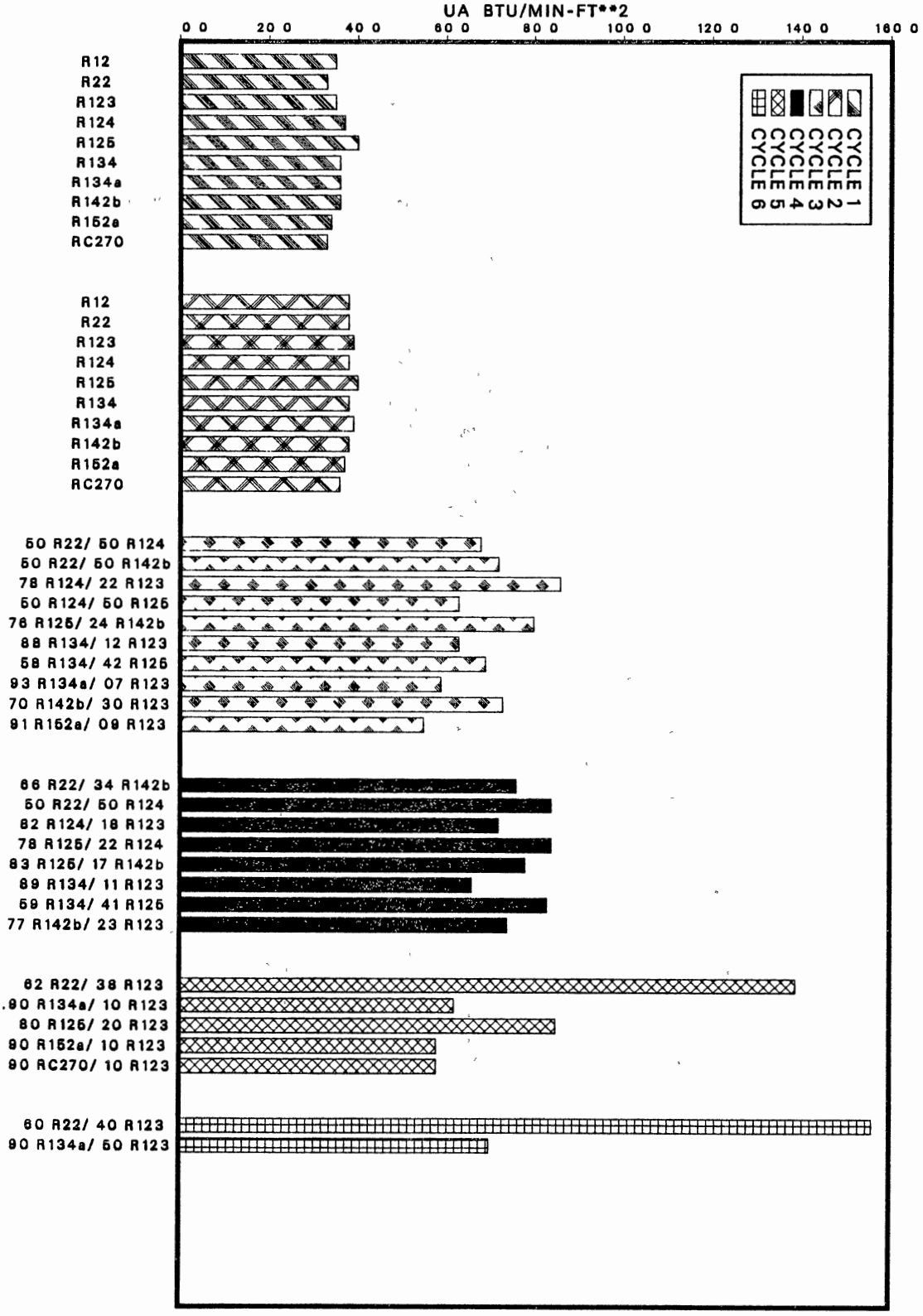


Figure 74. 15°F Source Temperature Loss- Total Heat Exchanger Conductance

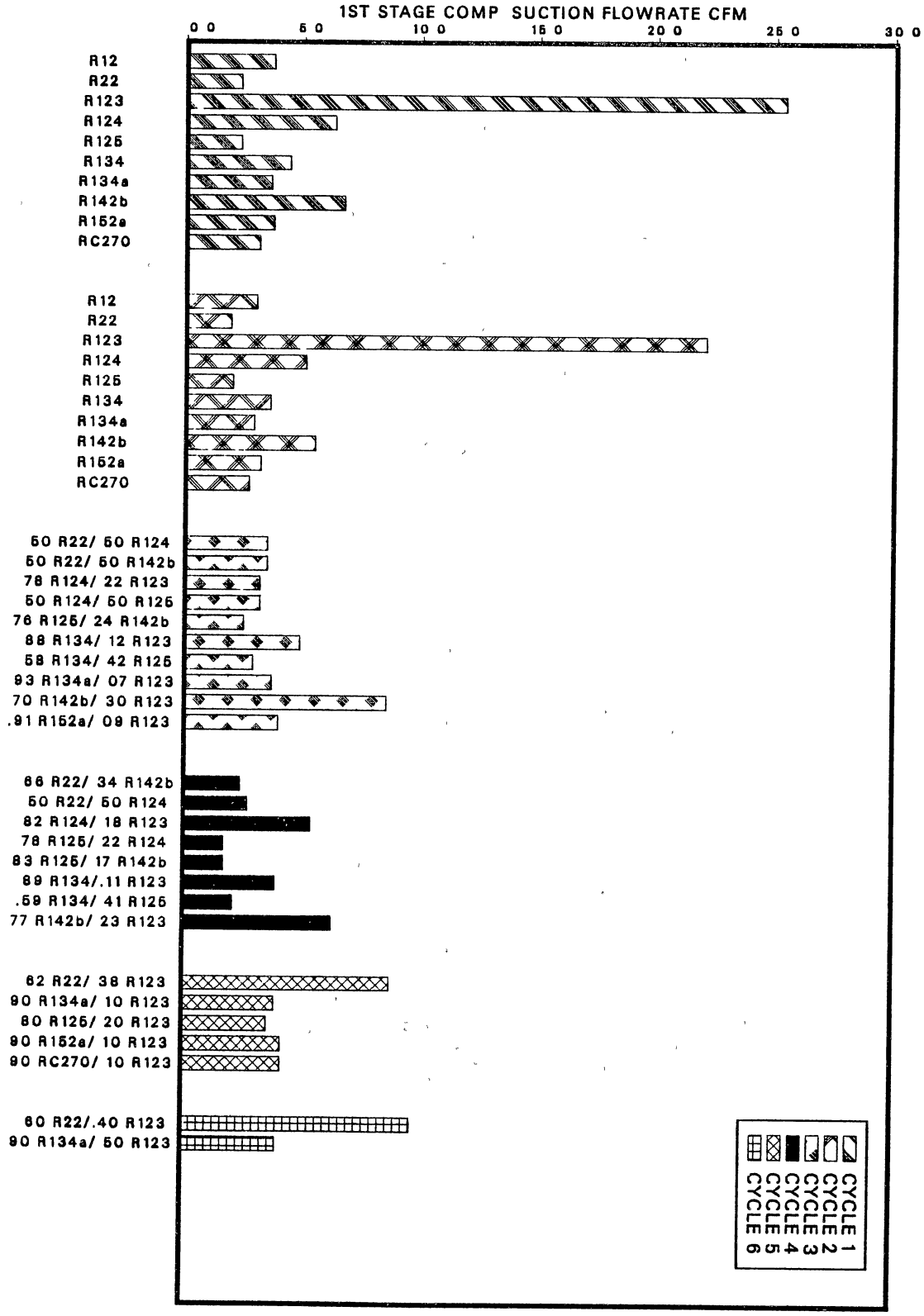


Figure 75. 15°F Source Temperature Loss- First Stage Compressor Volume Flow Rate

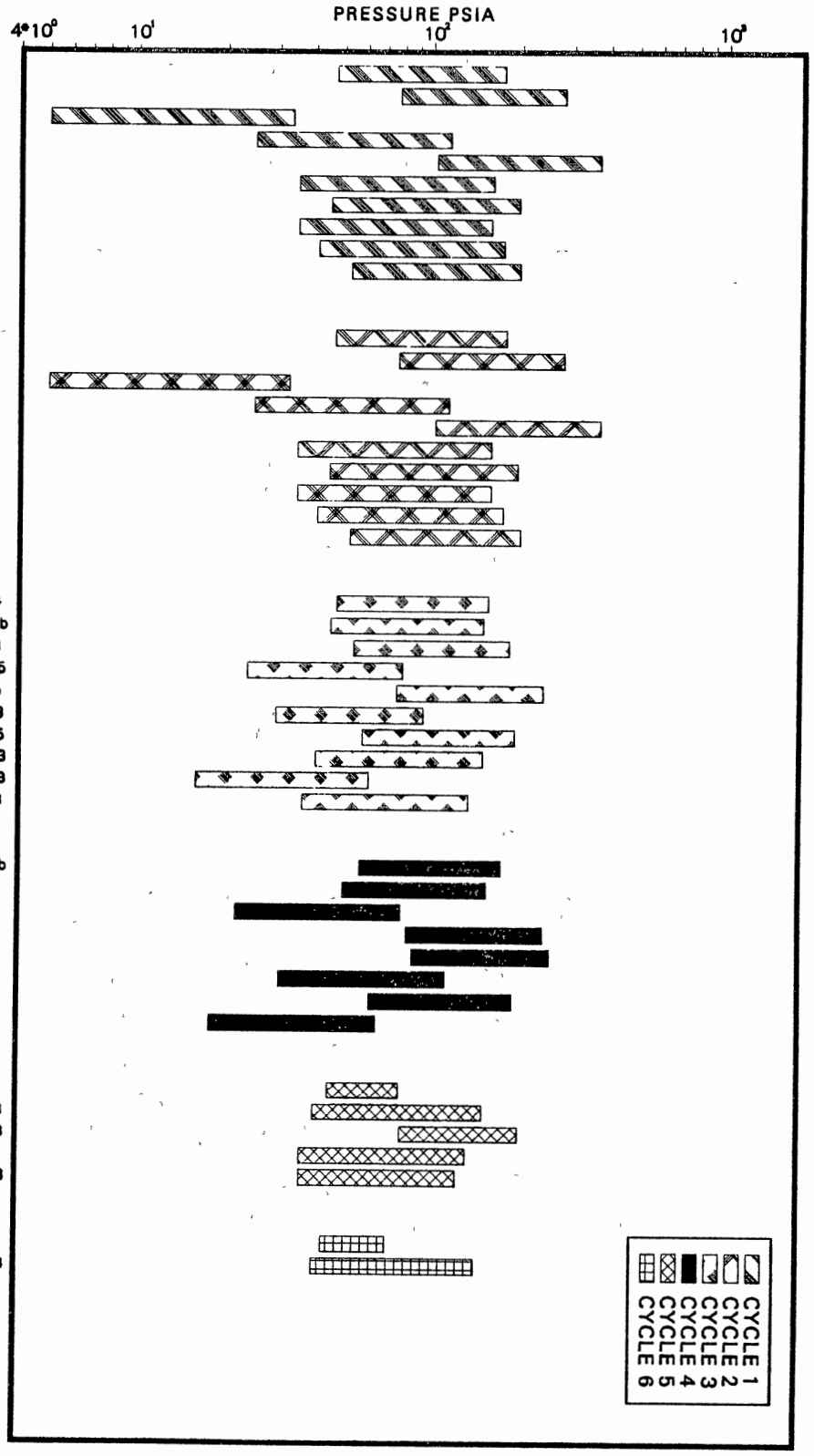


Figure 76. 15°F Source Temperature Loss- Operating Pressure Range

Similarly to Case I, for single-stage compressor systems, the performance and characteristics of NARM Cycle 3 were compared with those of pure refrigerant systems with Cycle 1.

The average of the improvement of the COPT for the optimum NARM's with Cycle 3 is approximately 11% higher than the pure refrigerant system with Cycle 1.

A comparison of the compressor pressure ratio shows the pressure ratio for the optimum NARM's with Cycle 3 is approximately 14% less than those for pure refrigerant systems.

The COPT's and improvement of the best NARM's for Cycle 3 compared to the pure refrigerant systems are given in Table XXXIII.

The compressor pressure ratio and pressure ratio reduction of the optimum NARM's for Cycle 3 compared to the pure refrigerant systems are given in Table XXXIV.

TABLE XXXIII

COPT IMPROVEMENT, NARM CYCLE 3 COMPARED TO CYCLE 1

	COPT			Percent Improvement	
		Pure Refrig.			
	Mixture	Ref. 1	Ref. 2	Ref.1	Ref.2
.50 R22/.50 R124	2.38	2.22	2.02	7.2	17.8
.50 R22/.50 R142b	2.56	2.22	2.25	15.3	13.8
.78 R124/.22 R123	2.38	2.02	2.10	17.8	13.3
.50 R124/.50 R125	2.32	2.02	1.91	14.9	21.5
.76 R125/.24 R142b	2.31	1.91	2.25	20.9	2.7
.88 R134/.12 R123	2.50	2.21	2.10	13.1	19.0
.58 R134/.42 R125	2.20	2.21	1.91	-0.5	15.2
.93 R134a/.07 R123	2.46	2.19	2.10	12.3	17.1
.70 R142b/.30 R123	2.52	2.25	2.10	12.0	20.0
.91 R152a/.09 R123	2.55	2.26	2.10	12.8	21.4

TABLE XXXIV

PRESSURE RATIO REDUCTION, NARM CYCLE 3 COMPARED TO CYCLE 1

	Pressure Ratio			Percent Reduction	
	Mixture	Pure Refrig. Ref. 1	Ref. 2	Ref.1	Ref.2
.50 R22/.50 R124	3.28	3.63	4.58	-9.6	-28.4
.50 R22/.50 R142b	3.30	3.63	4.51	-9.1	-26.8
.78 R124/.22 R123	3.38	4.58	6.65	-26.2	-49.2
.50 R124/.50 R125	3.36	4.58	3.58	-26.6	-6.1
.76 R125/.24 R142b	3.15	3.58	4.51	-12.0	-30.2
.88 R134/.12 R123	3.17	4.58	6.65	-30.8	-52.3
.58 R134/.42 R125	3.30	4.58	3.58	-27.9	-7.8
.93 R134a/.07 R123	3.70	4.36	6.65	-15.1	-44.4
.70 R142b/.30 R123	3.88	4.51	6.65	-14.0	-41.7
.91 R152a/.09 R123	3.67	4.27	6.65	-14.1	-44.8

Similarly to Case I, for two-stage compressor systems, the performance and characteristics of NARM Cycle 4 were compared with those of pure refrigerant systems with Cycle 2.

The average improvement of the COPT for the optimum NARM's with Cycle 4 is approximately 15.0% higher than the pure refrigerant system. The comparison of compressor pressure ratio shows that the pressure ratio for the optimum NARM's with Cycle 4 is approximately 17.0% less than those for pure refrigerant systems.

The COPT's and improvement of the optimum NARM's for Cycle 4 compared to pure refrigerant systems are given in Table XXXV.

The compressor pressure ratio and pressure ratio reduction of the optimum NARM's for Cycle 4 compared to pure refrigerant systems are given in Table XXXVI.

TABLE XXXV

COPT IMPROVEMENT, NARM CYCLE 4 COMPARED TO CYCLE 1

	COPT			Percent Improvement	
	Mixture	Pure Refrig. Ref. 1	Ref. 2	Ref.1	Ref.2
.66 R22/.34 R142b	2.61	2.22	2.25	17.6	16.0
.50 R22/.50 R124	2.61	2.22	2.02	17.6	29.2
.82 R124/.18 R123	2.44	2.02	2.10	20.8	16.2
.78 R125/.22 R124	2.31	1.91	2.02	20.9	14.4
.83 R125/.17 R142b	2.43	1.91	2.25	27.2	8.0
.89 R134/.11 R123	2.59	2.21	2.10	17.2	23.3
.94 R134a/.06 R123	2.54	2.19	2.10	16.0	21.0
.59 R134/.41 R125	2.30	2.21	1.91	4.1	20.4
.77 R142b/.23 R123	2.66	2.10	2.25	26.7	18.2

TABLE XXXVI

PRESSURE RATIO REDUCTION, NARM CYCLE 4 COMPARED TO CYCLE 1

Cycle 4 Mixtures	Pressure Ratio			Percent Reduction	
	Mixture	Pure Refrig. Ref. 1	Ref. 2	Ref.1	Ref.2
.66 R22/.34 R142b	3.02	3.63	4.51	-16.8	-33.0
.50 R22/.50 R124	3.08	3.63	4.58	-15.2	-32.8
.82 R124/.18 R123	3.66	4.58	6.65	-20.1	-45.0
.78 R125/.22 R124	2.90	3.58	4.58	-19.0	-36.7
.83 R125/.17 R142b	2.93	3.58	4.51	-18.2	-35.0
.89 R134/.11 R123	3.69	4.58	6.65	-19.4	-44.5
.94 R134a/.06 R123	3.67	4.36	6.65	-15.8	-44.8
.59 R134/.41 R125	3.06	4.58	3.58	-33.2	-14.5
.77 R142b/.23 R123	3.69	6.65	4.51	-44.5	-18.2

The performance and characteristics of NARM Cycle 5, (single-stage compressor) were compared with those of pure refrigerant systems with Cycle 1 (single-stage compressor).

The improvement of the COPT for the optimum NARM's with Cycle 5 is approximately 16.0% higher than the COPT for the pure refrigerant system. Note that not all NARM system have better performance than pure systems.

The comparison of the compressor pressure ratio shows that the pressure ratio for the optimum NARM's with Cycle 5 is approximately 24.0% less than those for pure refrigerant systems.

The COPT's and improvement of the optimum NARM's for Cycle 3 compared to the pure refrigerant systems are given in Table XXXVII.

TABLE XXXVII

COPT IMPROVEMENT, NARM CYCLE 5 COMPARED TO CYCLE 1

Cycle 5 Mixtures	COPT			Percent Improvement	
		Pure Refrig.			
	Mixture	Ref. 1	Ref. 2	Ref.1	Ref.2
.62 R22/.38 R123	2.44	2.22	2.10	9.9	16.2
.90 R134a/.10 R123	2.55	2.19	2.10	16.4	21.4
.80 R125/.20 R123	2.20	1.91	2.10	15.2	4.8
.90 R152a/.10 R123	2.53	2.26	2.10	11.9	20.5
.90 RC270/.10 R123	2.68	2.10	2.29	27.6	17.0

The compressor pressure ratio and pressure ratio reduction of the optimum NARM's for Cycle 5 compared to the pure refrigerant systems are given in Table XXXVIII.

TABLE XXXVIII

PRESSURE RATIO REDUCTION, NARM CYCLE 5 COMPARED TO CYCLE 1

Cycle 5 Mixtures	Pressure Ratio			Percent Reduction	
	Mixture	Pure Refrig. Ref. 1	Ref. 2	Ref.1	Ref.2
.62 R22/.38 R123	1.74	3.63	6.65	-52.1	-73.8
.90 R134a/.10 R123	3.76	4.36	6.65	-13.8	-43.5
.80 R125/.20 R123	2.52	3.58	6.65	-29.6	-62.1
.90 R152a/.10 R123	3.67	4.27	6.65	-14.1	-44.8
.90 RC270/.10 R123	3.40	6.65	3.74	-48.9	-9.1

For a low source side temperature loss system. The comparison between NARM Cycle 5 and NARM Cycle 6 shows that the performance and characteristics of Cycle 5 and Cycle 6 were not much difference.

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

Models of six vapor compression cycles were constructed to identify and evaluate the performance of vapor compression cycles operating with acceptable non-azeotropic refrigerant mixtures. The temperature range considered in this study was for an air conditioner and chiller system application. The refrigerant and refrigerant mixture properties were based on the CSD equation of state and the program prepared by NIST was used.

Two operating conditions were evaluated to compare the performance of NARM systems to pure refrigerant systems. The performance was based on a one ton cooling load for an air cooled condenser system. The total electric power consumption used to evaluate the performance is the sum of the power consumption for the compressor and for the condenser fan.

The six vapor compression cycles were:

- Two cycles for pure refrigerants (used for comparison purpose only).
- Two cycles for NARM with single phase vapor at the outlet of the evaporator.
- Two cycles for NARM with two-phase condition at the evaporator outlet.

The two operating conditions considered in this study are discussed in section 6.1 and 6.2.

6.1 Operating Condition I

The heat sink temperature was set at 100°F, the heat source inlet temperature at 70°F with a 30°F heat source temperature drop.

The improvements of the individual NARM and cycles compared to the pure refrigerant cycles were listed in Section 5.2 (Tables XVII, XIX, XXI, XXIII, and XXV). The reduction of the compressor pressure ratio of the NARM cycles compared to the pure refrigerant systems were listed in Tables XVIII, XX, XXII, XXIV and XXVI. The average of total heat exchanger conductances for each cycle were given in Table XXVII.

In general, the results of the simulation indicated that :

- Cycle 5 and Cycle 6 were the heaviest systems (biggest heat exchanger conductance). However, the cycles have very good performance as well as the lowest compressor pressure ratio.

- The Cycle 4 had the highest COPT and was in the medium range of weight and pressure ratio.

- The coefficient of performance of the advanced NARM cycles (Cycle 4, Cycle 5 and Cycle 6) is approximately 7-8% higher than those of the basic NARM cycle (Cycle 3).

- Binary non-azeotropic mixtures with R-123 were found frequently in the optimum NARM's. Refrigerant R-123 were the most common constituent of NARM's.

- The summary of the performance improvement, compressor pressure ratio, and heat exchanger sizes is given in Table XXXIX.

- Some typical Non-azeotropic mixtures for each cycle are summarized in Table XXXX.

TABLE XXXIX

COMPARISON OF PERFORMANCE, PRESSURE RATIO, AND HEAT EXCHANGER CONDUCTANCE BETWEEN NARM AND PURE REFRIGERANT SYSTEM

Cycles	Percent Performance Improvement	Percent Comp. Press. Ratio Reduction	Percent UA Increment
Cycle 3	20.	-7.	90.
Cycle 4	30.	-14.	103.
Cycle 5	28.	-30.	206.
Cycle 6	28.	-31.	227.

TABLE XXXX

TYPICAL NARMS FOR EACH CYCLE

Cycle	Mixture 1	Mixture 2
Cycle 3	.78 R-134a/.22 R-123	.58 R-124/.42 R-123
Cycle 4	.82 R-134a/.18 R-123	.59 R-124/.41 R-123
Cycle 5	.60 R-22/.40 R-123	.72 R-125/.28 R-123
Cycle 6	.50 R-22/.50 R-123	.70 R-125/.30 R-123

6.2 Operating Condition II

The heat sink temperature was set at 100°F and the heat source inlet temperature at 55°F, with a 15°F heat source temperature drop.

The improvements of the individual NARM and cycles compared to the pure refrigerant cycles were detailed in Section 5.3 (Tables XXXIII, XXXV, and XXXVII). The reduction of the compressor pressure ratio of

the NARM cycles compared to the pure refrigerant systems were given in Tables XXXIV, XXXVI and XXXVIII.

In general, the results of the simulation indicated that :

- The lower the heat source temperature drop, the lower the improvement of NARM cycles over pure refrigerant cycles.

- The coefficient of performance of advanced NARM cycles (Cycle 4, Cycle 5 and Cycle 6) are approximately 3-4% higher than those of the basic NARM cycle (Cycle 3).

- The performance of Cycle 6 showed no gain from Cycle 5 and would therefore not be recommended for a low heat sink temperature drop system.

- Similarly to Condition I, binary non-azeotropic mixtures with R-123 were found frequently in the optimum NARM's. Refrigerant R-123 were the most common constituent of NARM's.

- A summary of the performance improvement and compressor pressure ratio is given in Table XXXXI.

TABLE XXXXI

COMPARISON OF PERFORMANCE, PRESSURE RATIO, AND HEAT EXCHANGER CONDUCTANCE BETWEEN NARM AND PURE REFRIGERANT SYSTEM

Cycles	Percent Performance Improvement	Percent Comp. Press. Ratio Reduction
Cycle 3	11.	-14.
Cycle 4	15.	-17.
Cycle 5	16.	-24.

- Some typical Non-azeotropic mixtures for each cycle are summarized in Table XXXXII.

TABLE XXXXII

TYPICAL NARMS FOR EACH CYCLE

Cycle	Mixture 1	Mixture 2
Cycle 3	.88 R-134/.12 R-123	.78 R-124/.22 R-123
Cycle 4	.89 R-134/.11 R-123	.82 R-124/.18 R-123
Cycle 5	.62 R-22/.38 R-123	.80 R-125/.20 R-123

6.3 Recommendations for Future Work

1. Depending on the type of compressor, the compressor inlet conditions may be required to be a saturated vapor or a superheated vapor. In this study, saturated vapor was assumed to exist at the inlet of the compressor. For centrifugal compressors, as an example, superheated vapor is required at the compressor inlet. This directly affects the temperature profile in the evaporator. The effect of the degree of superheat existing at the evaporator outlet should be evaluated relative to the performance of the vapor compression cycle for Cycle 3 and Cycle 4. For Cycle 5 and Cycle 6, superheating was done in the subcooler SUB 1. Therefore, the performance of Cycle 5 and Cycle 6 are affected very little by the evaporator outlet superheating requirements.

2. This study concentrated on applications to air conditioners and chiller systems with an air cooled condenser. In the future the performance of a vapor compression cycle with a liquid cooled condenser. The power consumption for condenser pump and cooling tower fan should be included in the evaluation of system performance.

3. Heat pump applications can benefit from the high temperature gradients that are available in the condensers of NARM vapor compression cycles. The heating coefficient of performance of NARM heat pumps for Cycle 4, Cycle 5, and Cycle 6 should be evaluated.

4. To gain the benefit of NARM cycles, The evaporator and condenser need to be designed to have a counterflow configuration. However, for an air source and air sink system, a complete counterflow evaporator or condenser is complicated. The designing of counter-flow heat exchangers for air-to-air system should be considered and create better heat exchanger models.

5. The line pressure drop should be included to the simulation and the simulation results should be compared to experimental results.

6. The effect of the compression pressure ratio to the efficiency of compressor.

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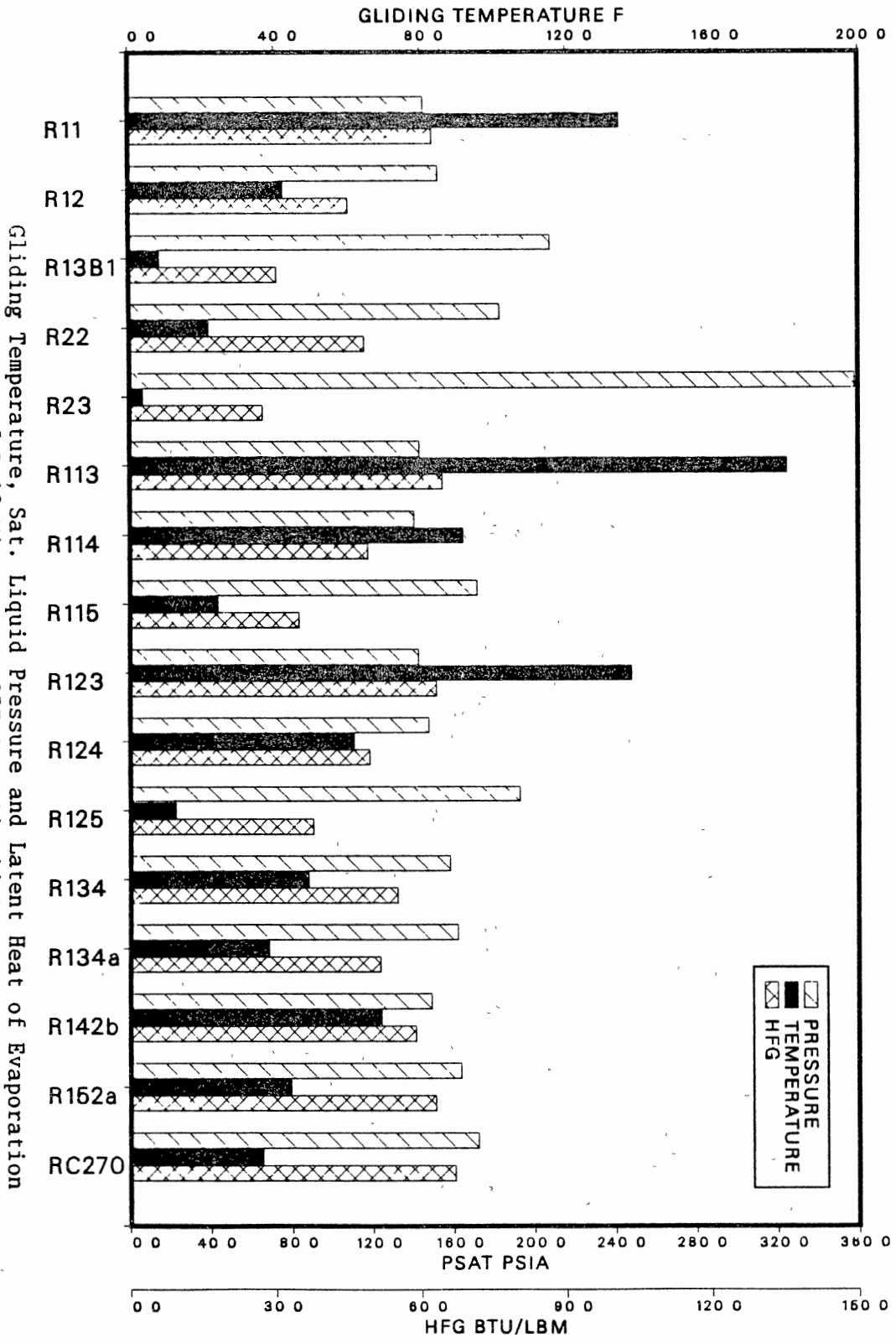
APPENDIX A

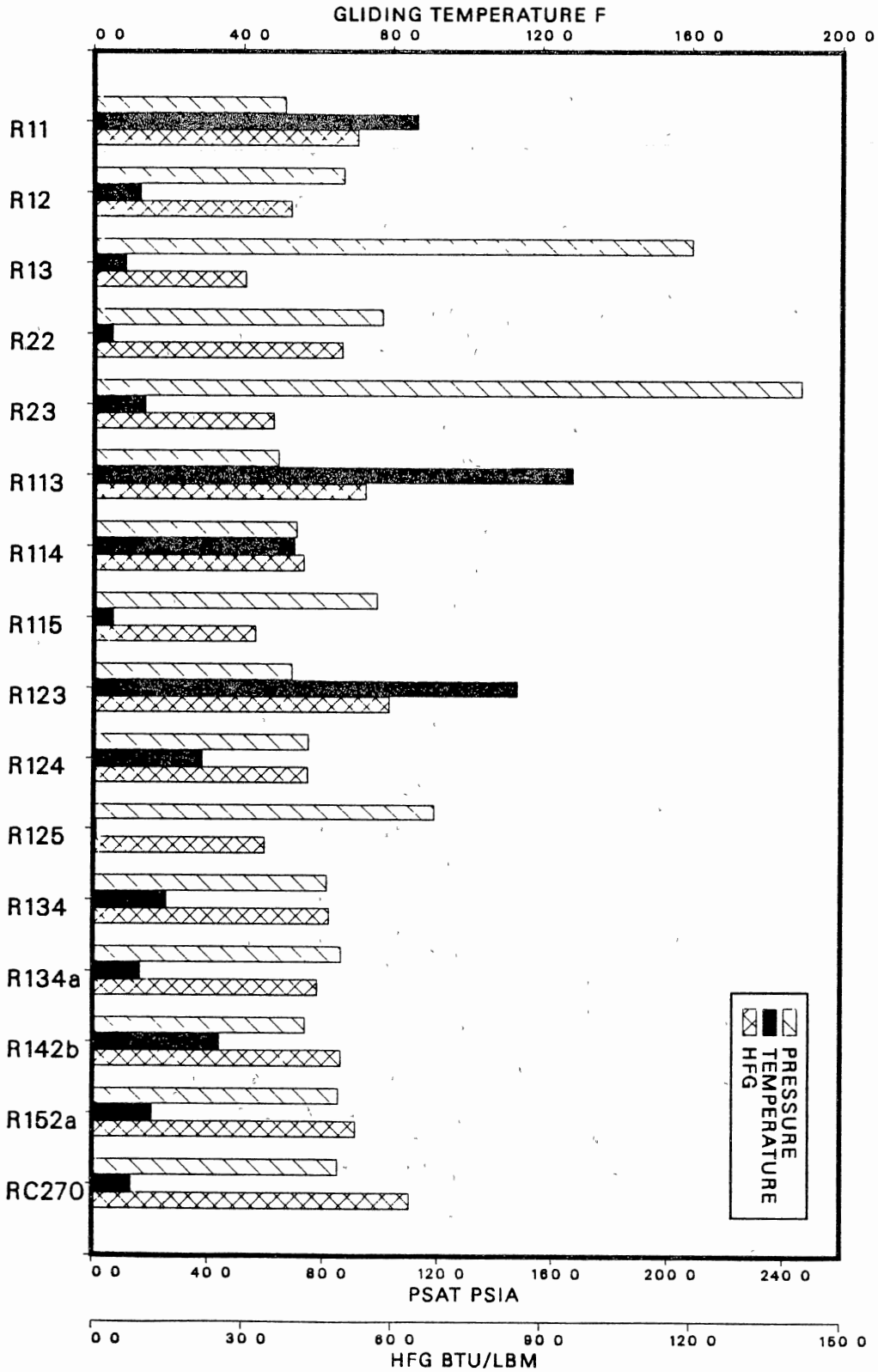
GLIDING TEMPERATURE, SATURATED LIQUID PRESSURE AND LATENT HEAT OF EVAPORATION OF NARMS

Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation
of R-11 Mixtures at 35F Sat. Liquid Pressure

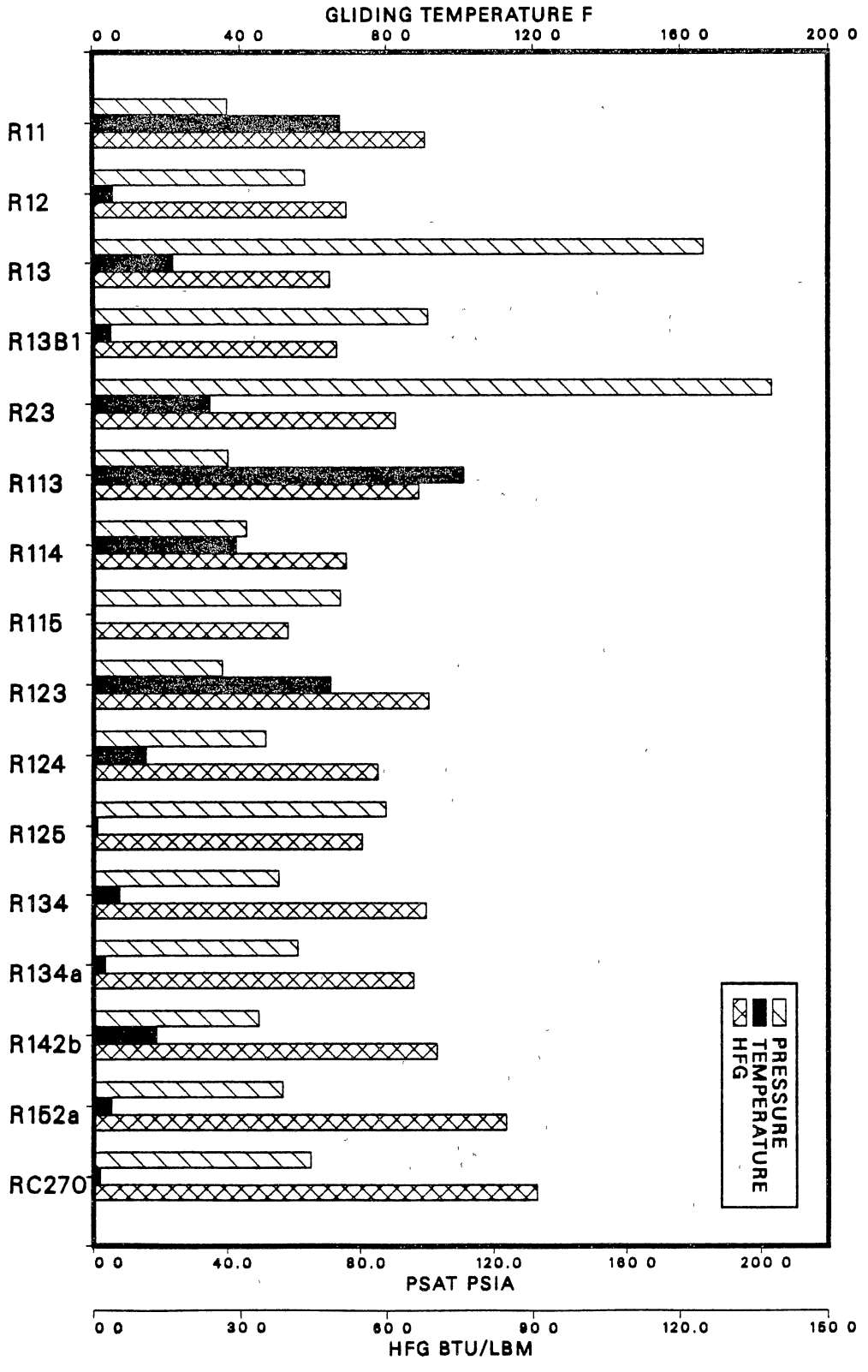
Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-12 Mixtures at 35F Sat. Liquid Pressure



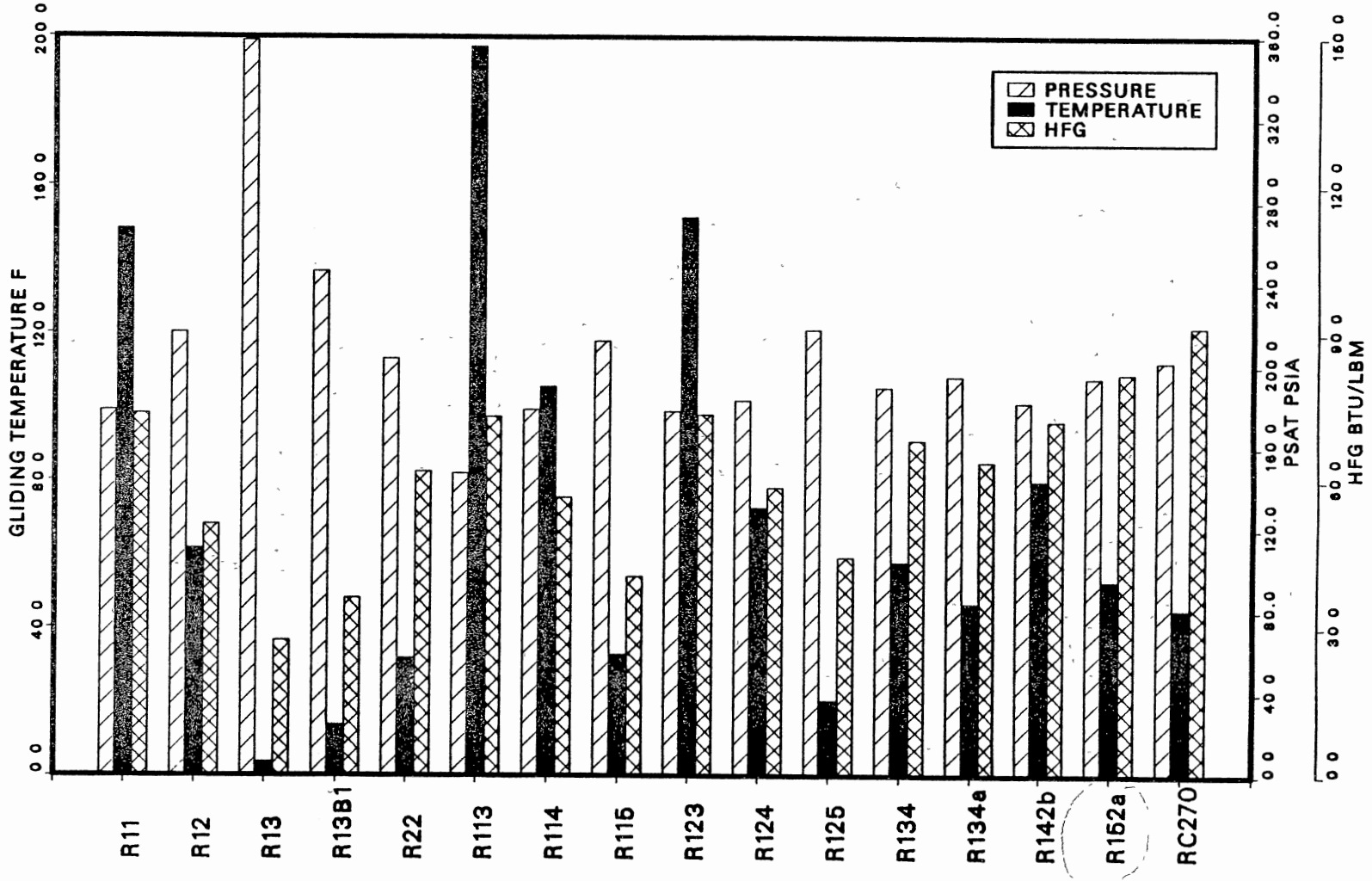
Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-13 Mixtures at 35F Sat. Liquid Pressure



Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-13B1 Mixtures at 35F Sat. Liquid Pressure

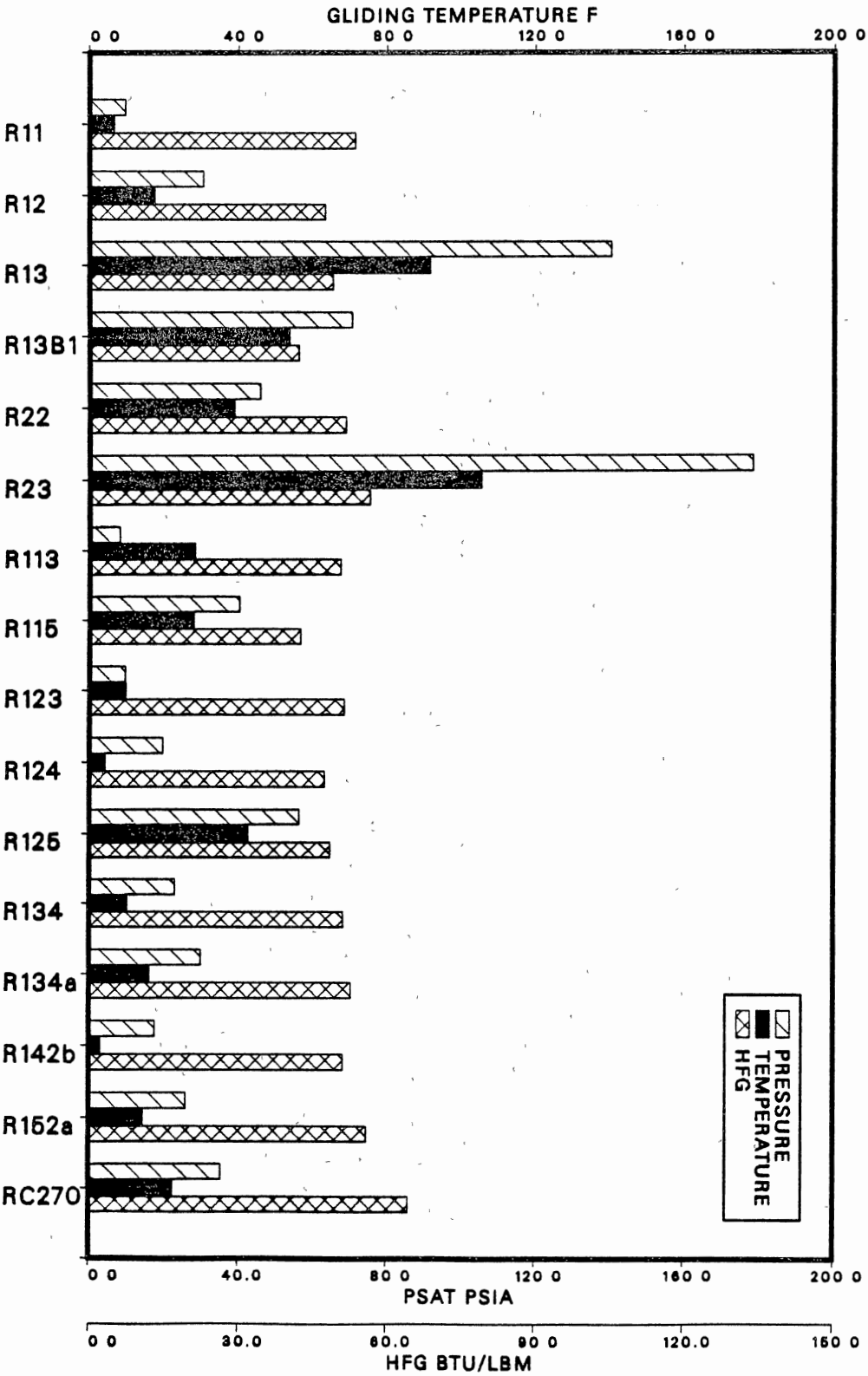


Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-22 Mixtures at 35F Sat. Liquid Pressure

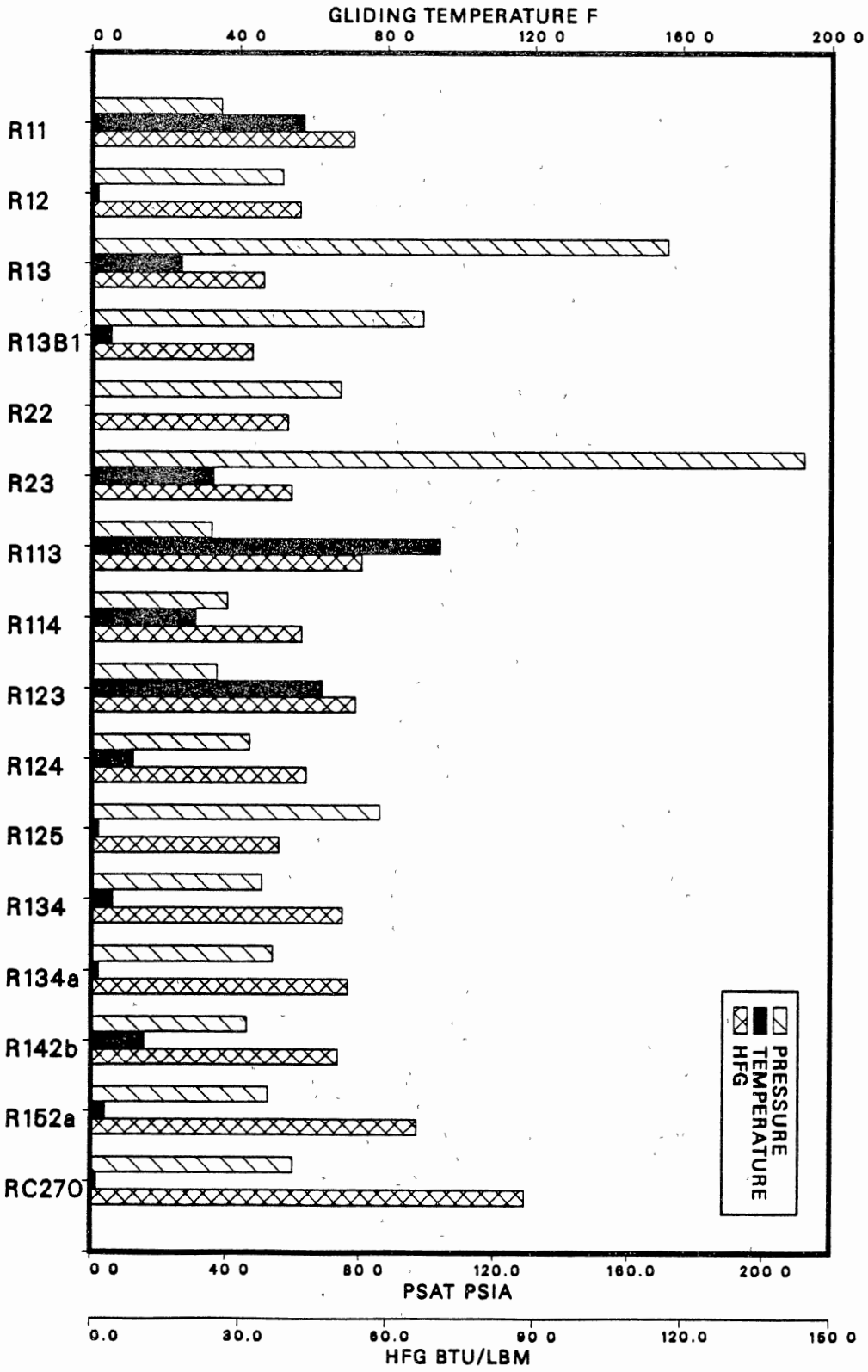


Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-23 Mixtures at 35F Sat. Liquid Pressure

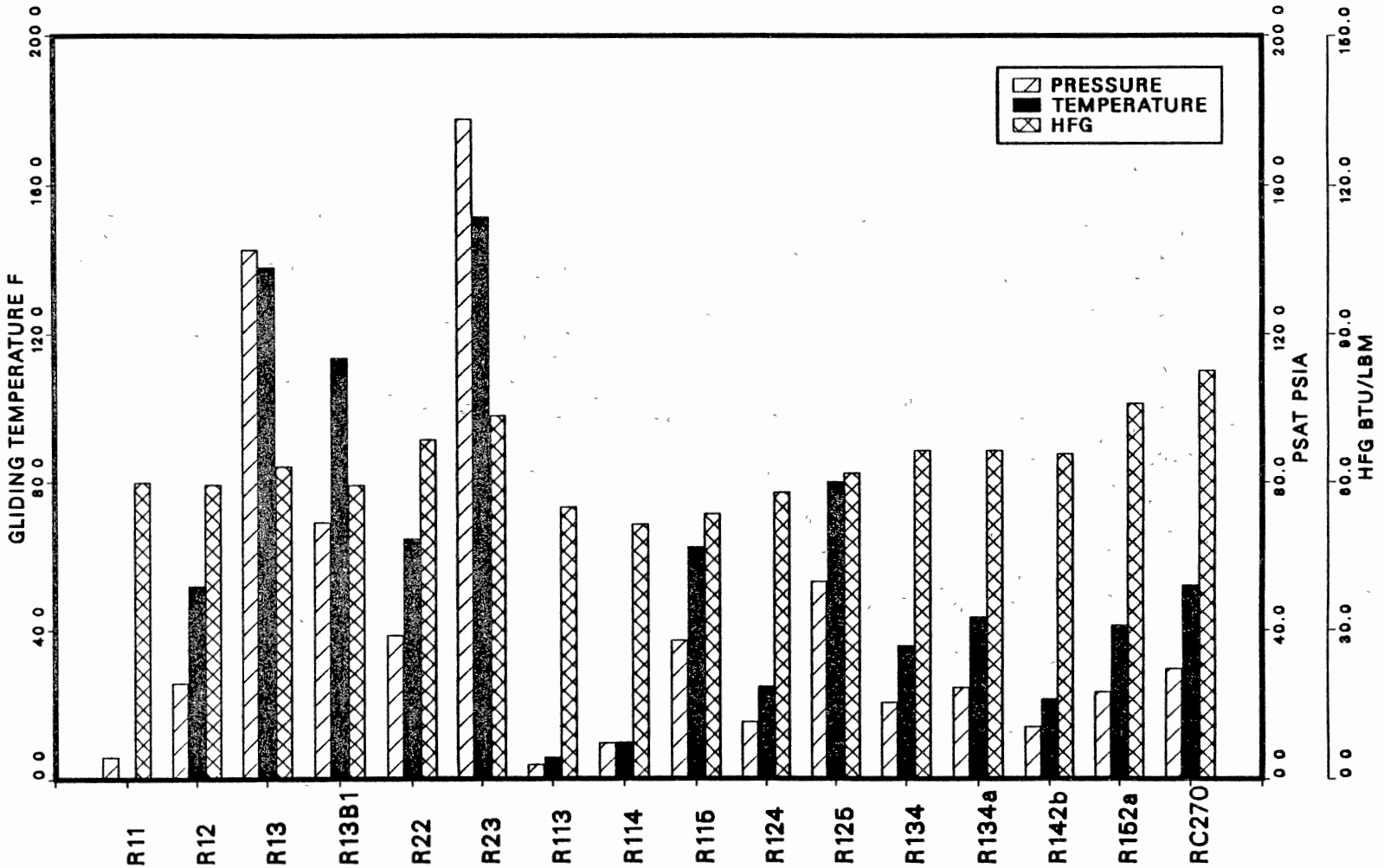
Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation
of R-113 Mixtures at 35F Sat. Liquid Pressure



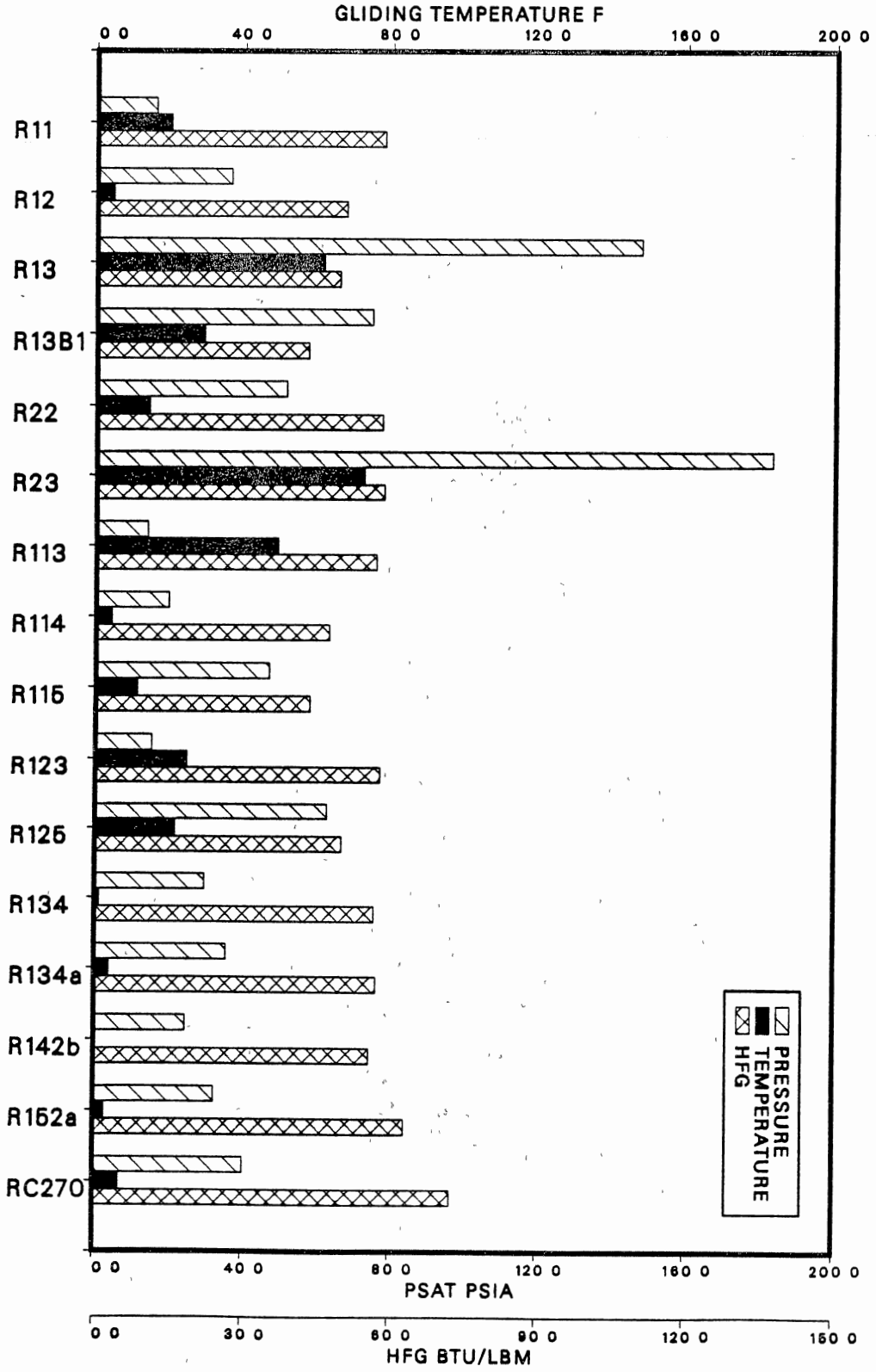
Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-114 Mixtures at 35F Sat. Liquid Pressure



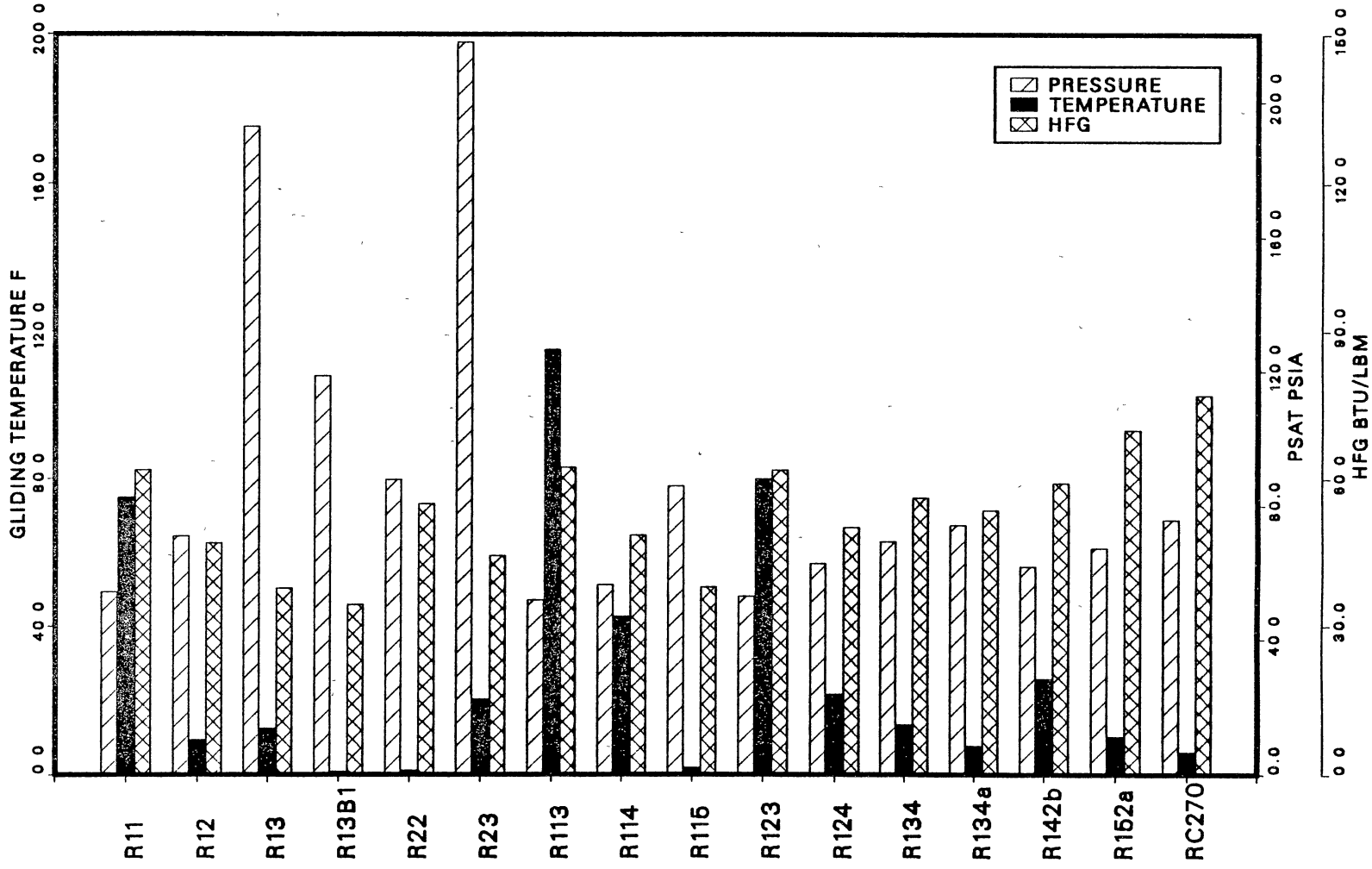
Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-115 Mixtures at 35F Sat. Liquid Pressure



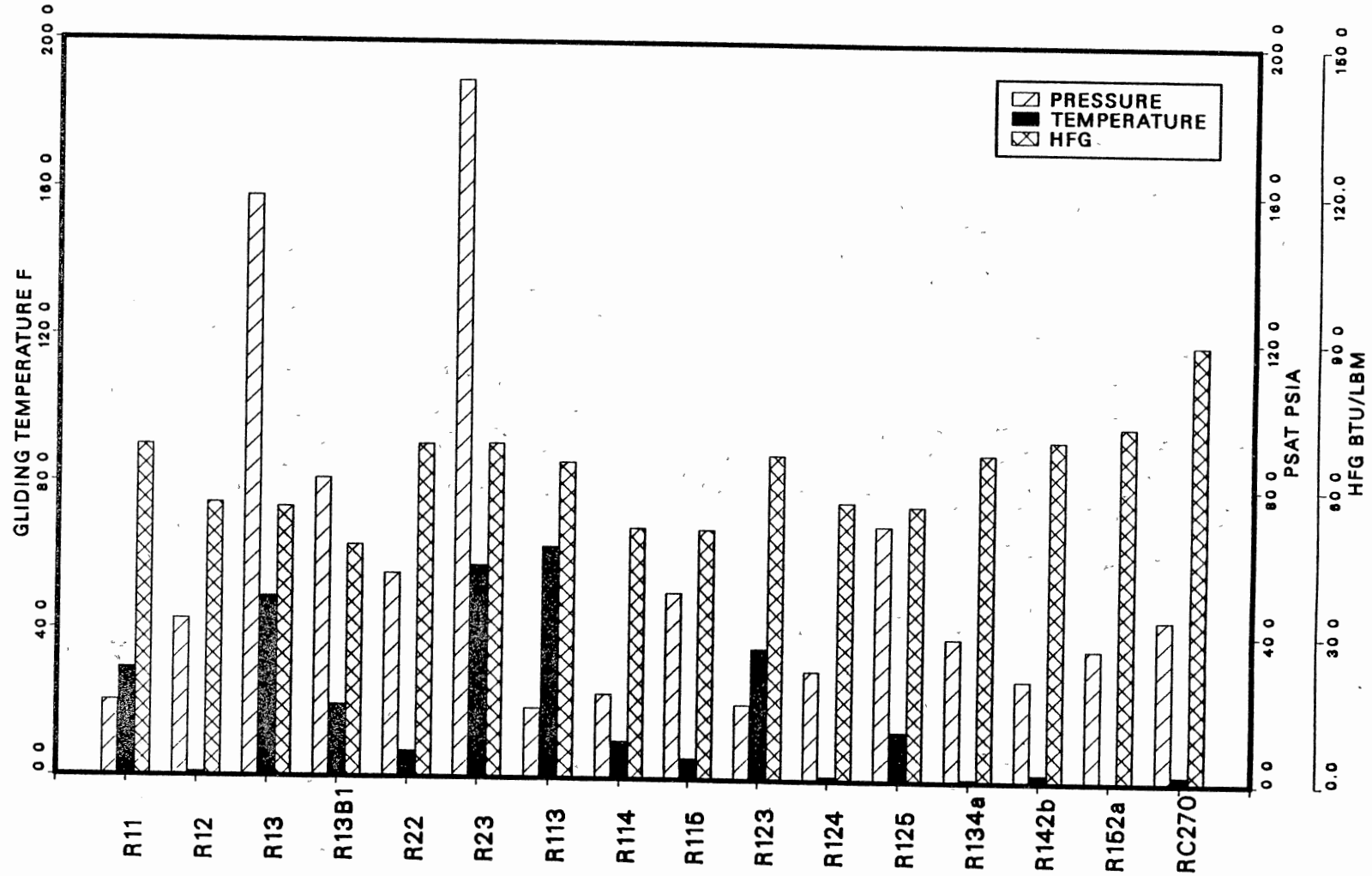
Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-123 Mixtures at 35F Sat. Liquid Pressure



Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-124 Mixtures at 35F Sat. Liquid Pressure

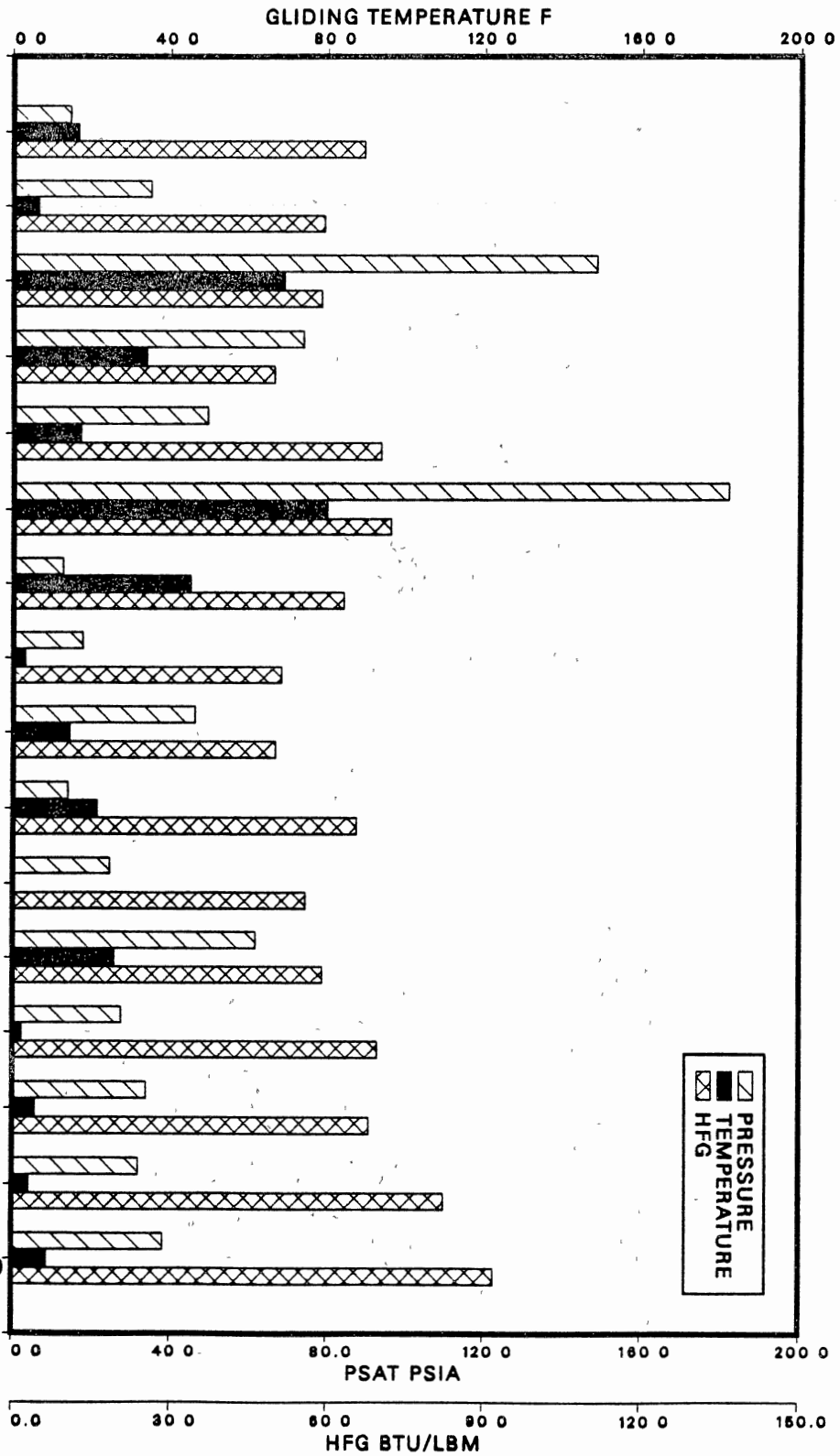


Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-125 Mixtures at 35F Sat. Liquid Pressure

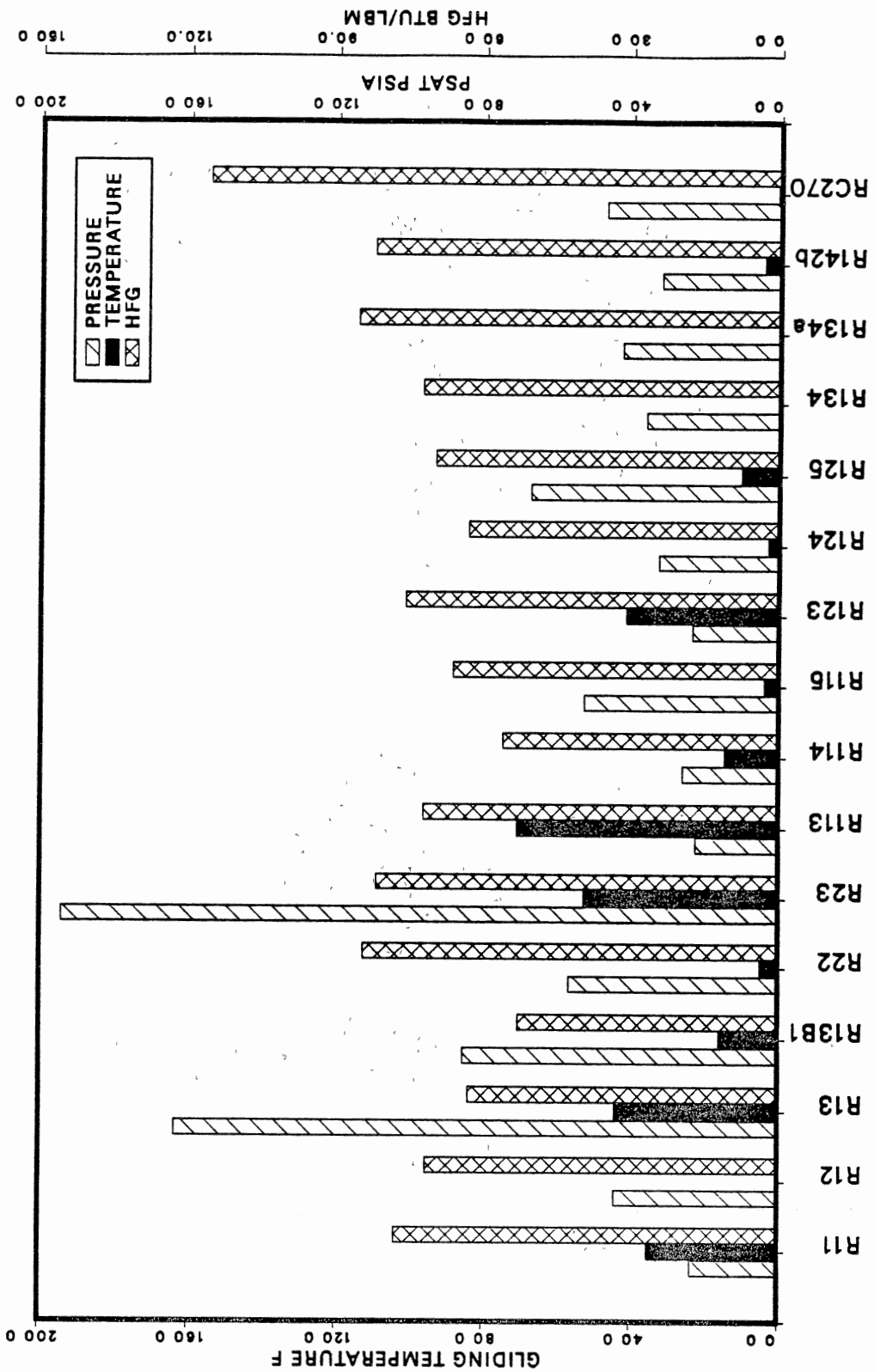


Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation
of R-134 Mixtures at 35F Sat. Liquid Pressure

Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-134a Mixtures at 35F Sat. Liquid Pressure



Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-142b Mixtures at 35F Sat. Liquid Pressure

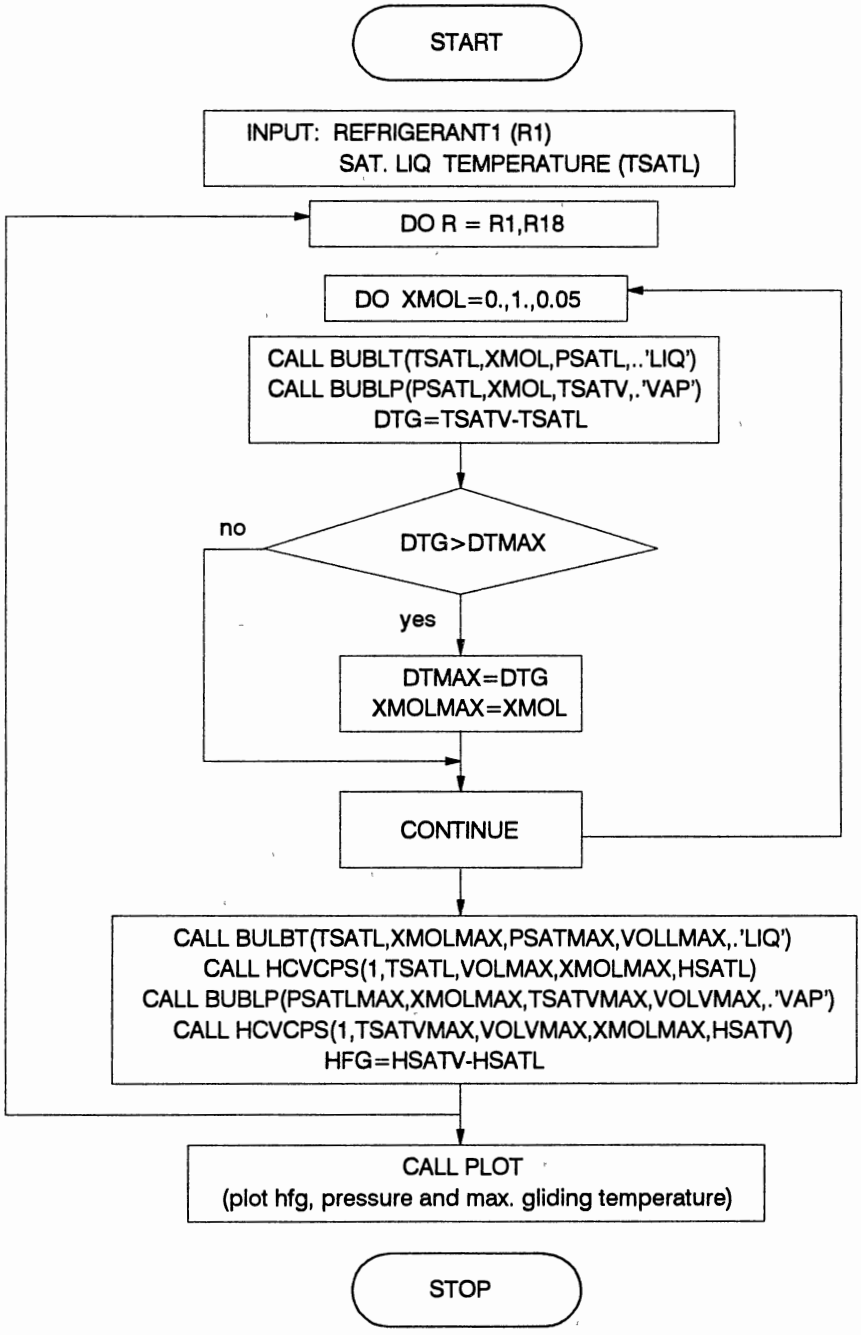


Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of R-152a Mixtures at 35F Sat. Liquid Pressure

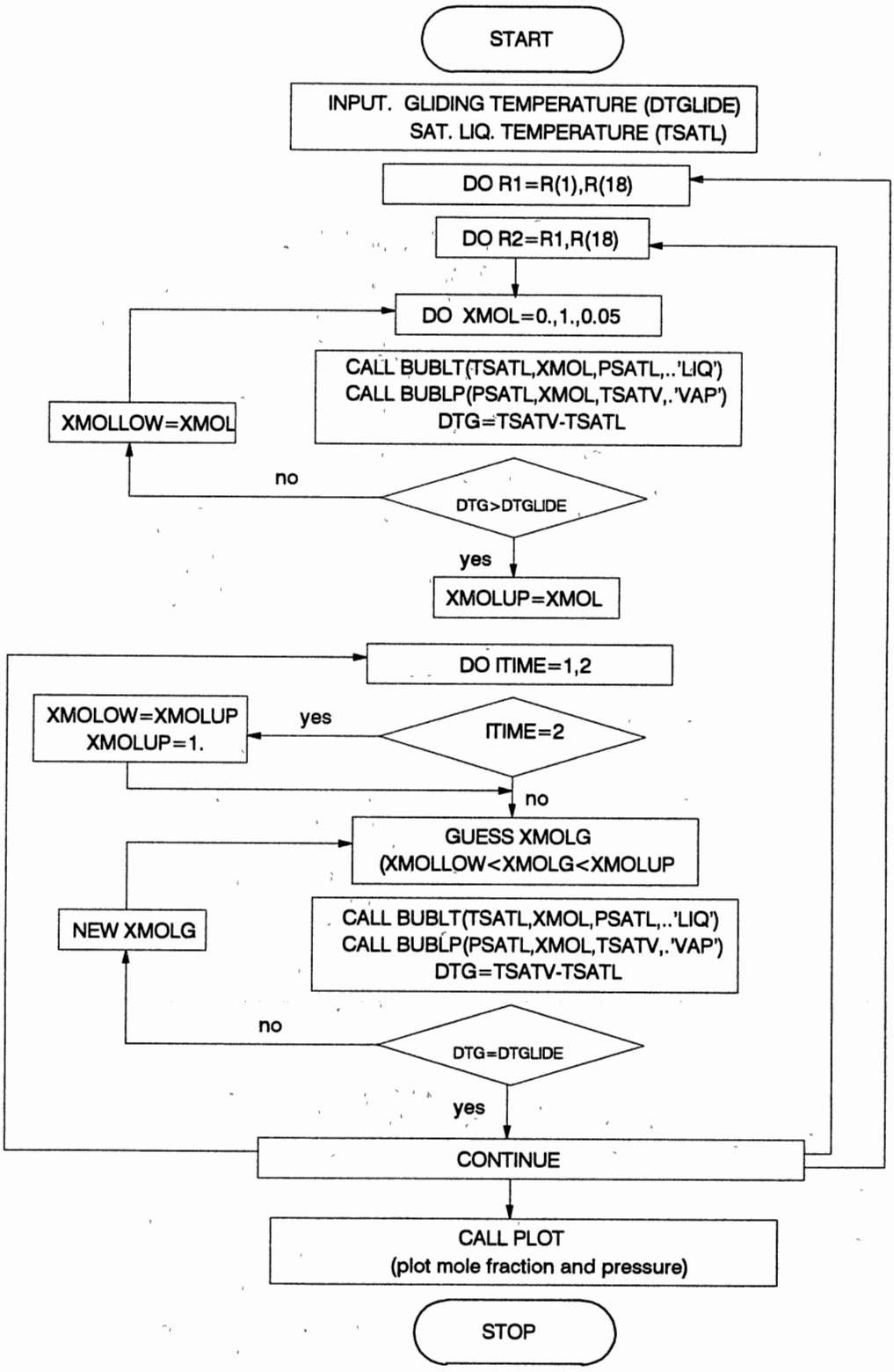
Gliding Temperature, Sat. Liquid Pressure and Latent Heat of Evaporation of RC-270 Mixtures at 35F Sat. Liquid Pressure

APPENDIX B

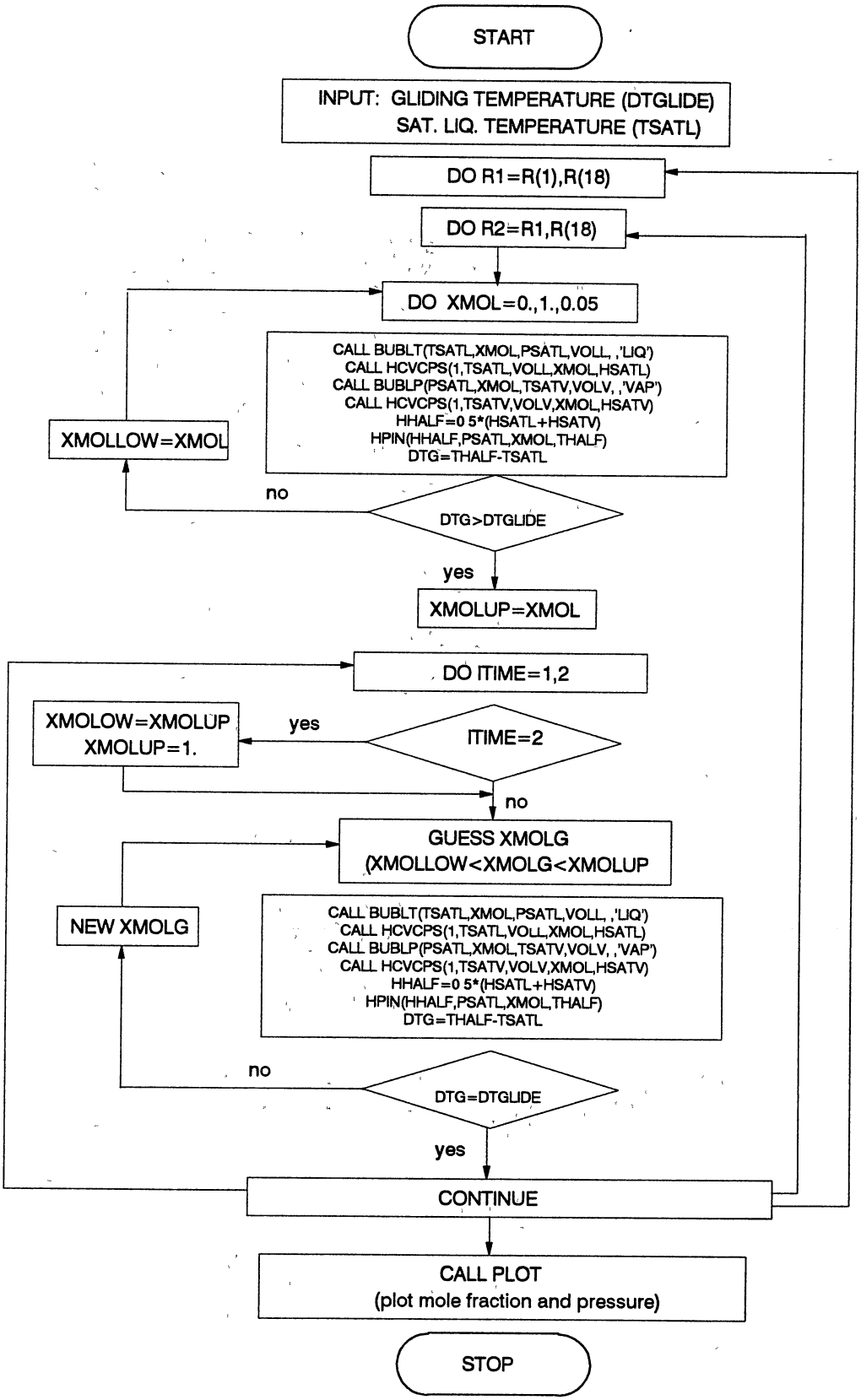
FLOW CHART TO GENERATE GLIDING TEMPERATURE INTERVALS AND SIMULATION MODEL FOR CYCLE 6



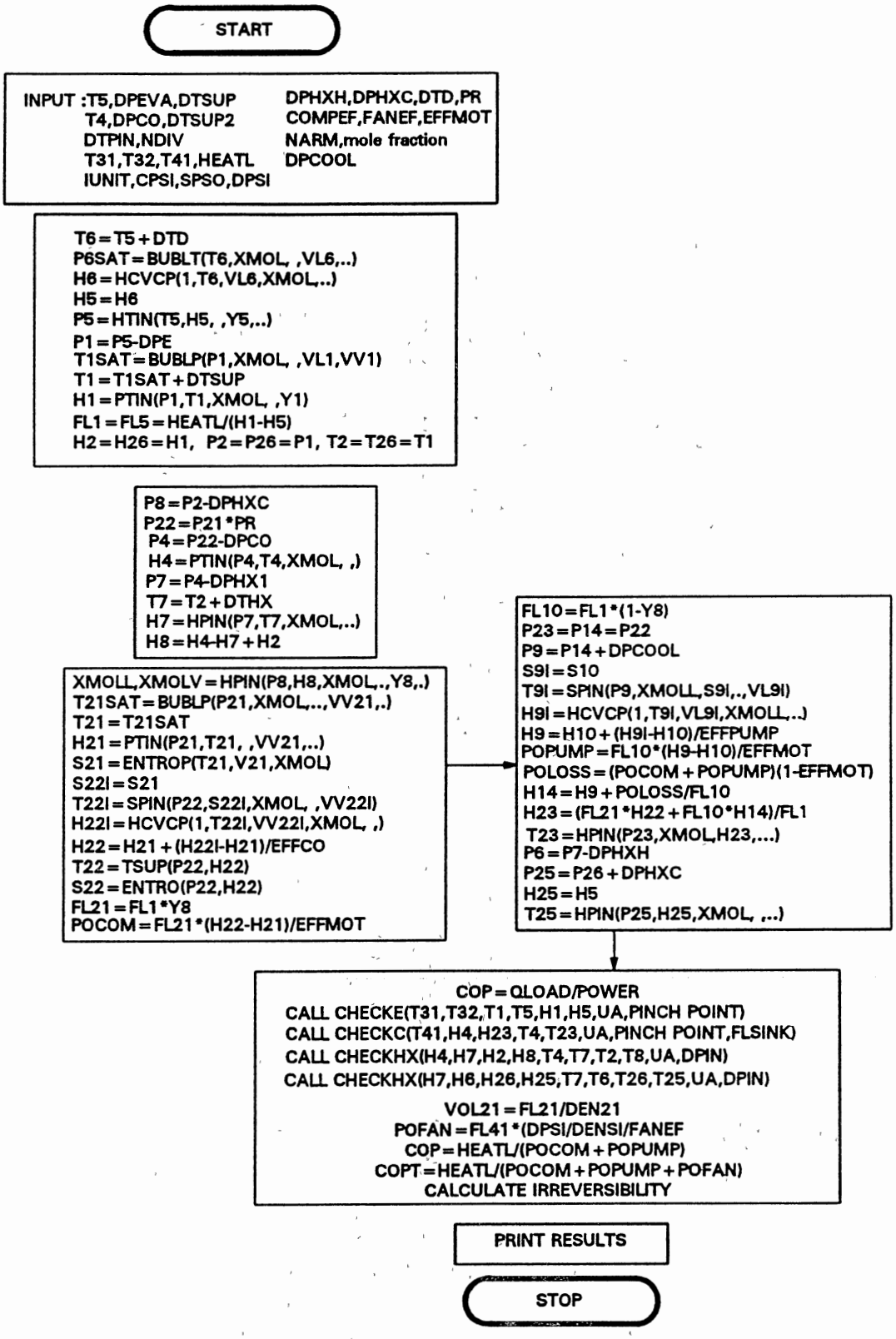
Flow Chart to Calculate Maximum Gliding Temperature



Flow Chart to Predict Mole Fraction of NARM for Cycle 3 and 4



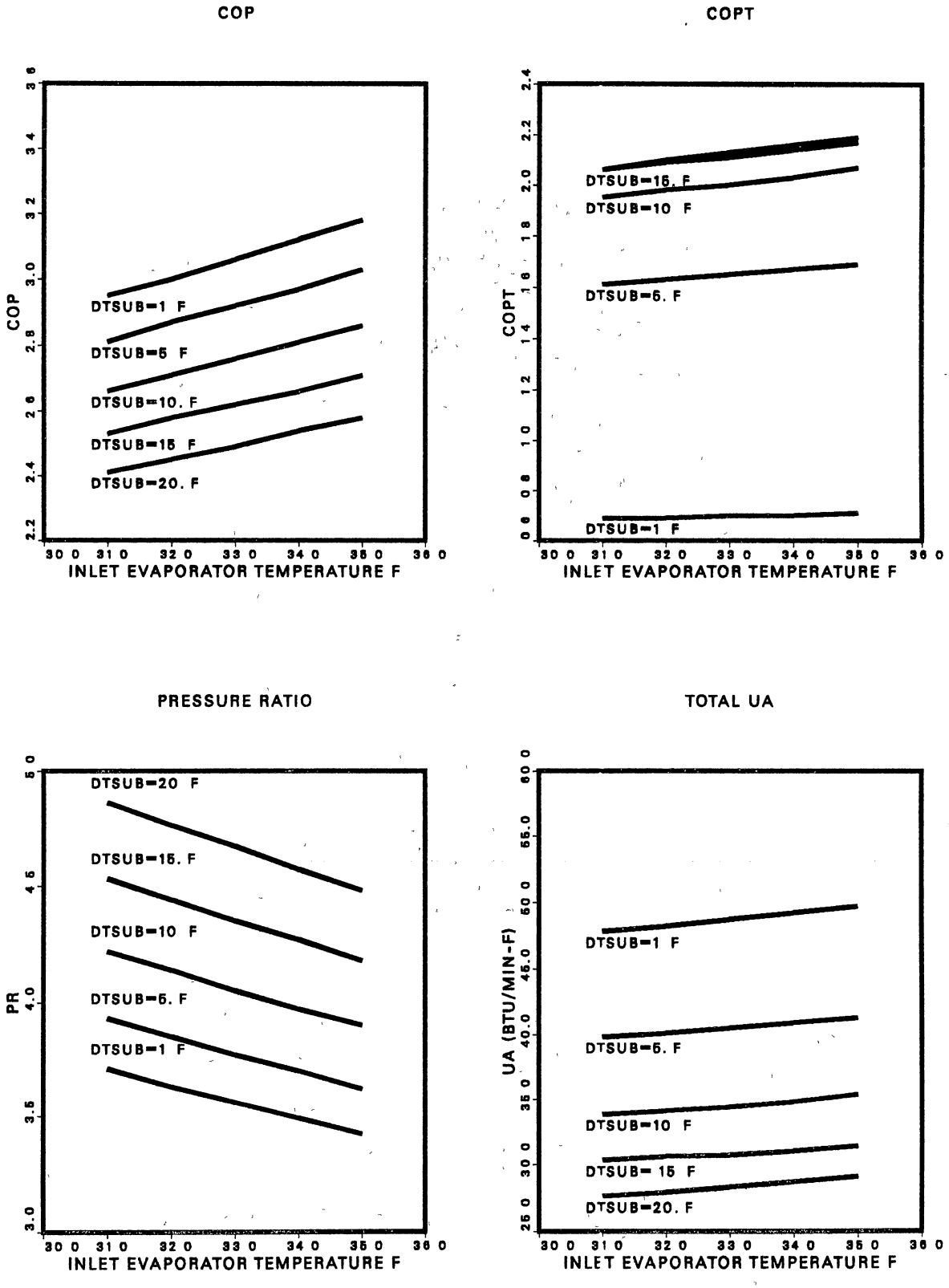
Flow Chart to Predict Mole Fraction of NARM for Cycle 5 and 6



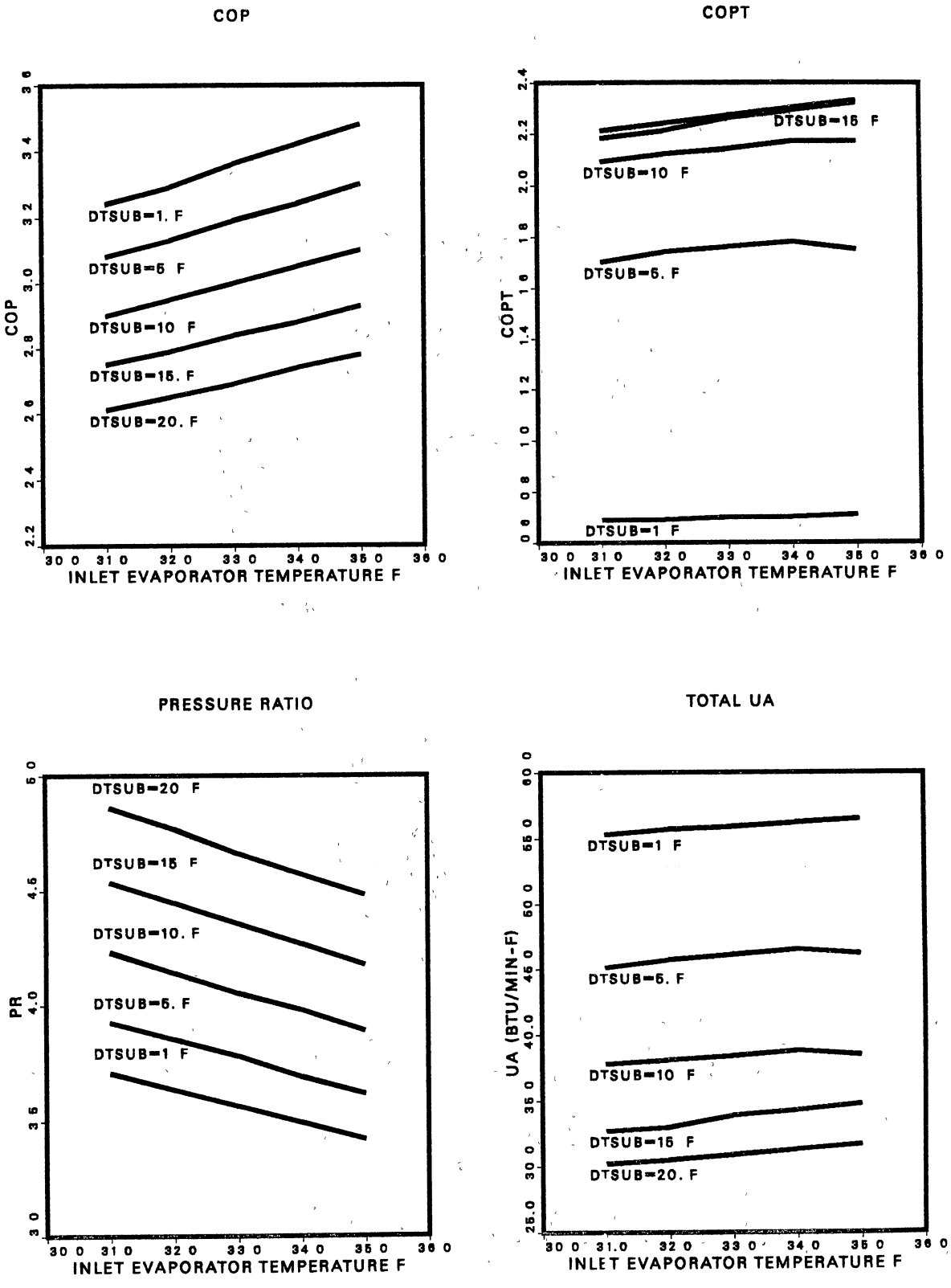
Flow Chart for Cycle 6

APPENDIX C

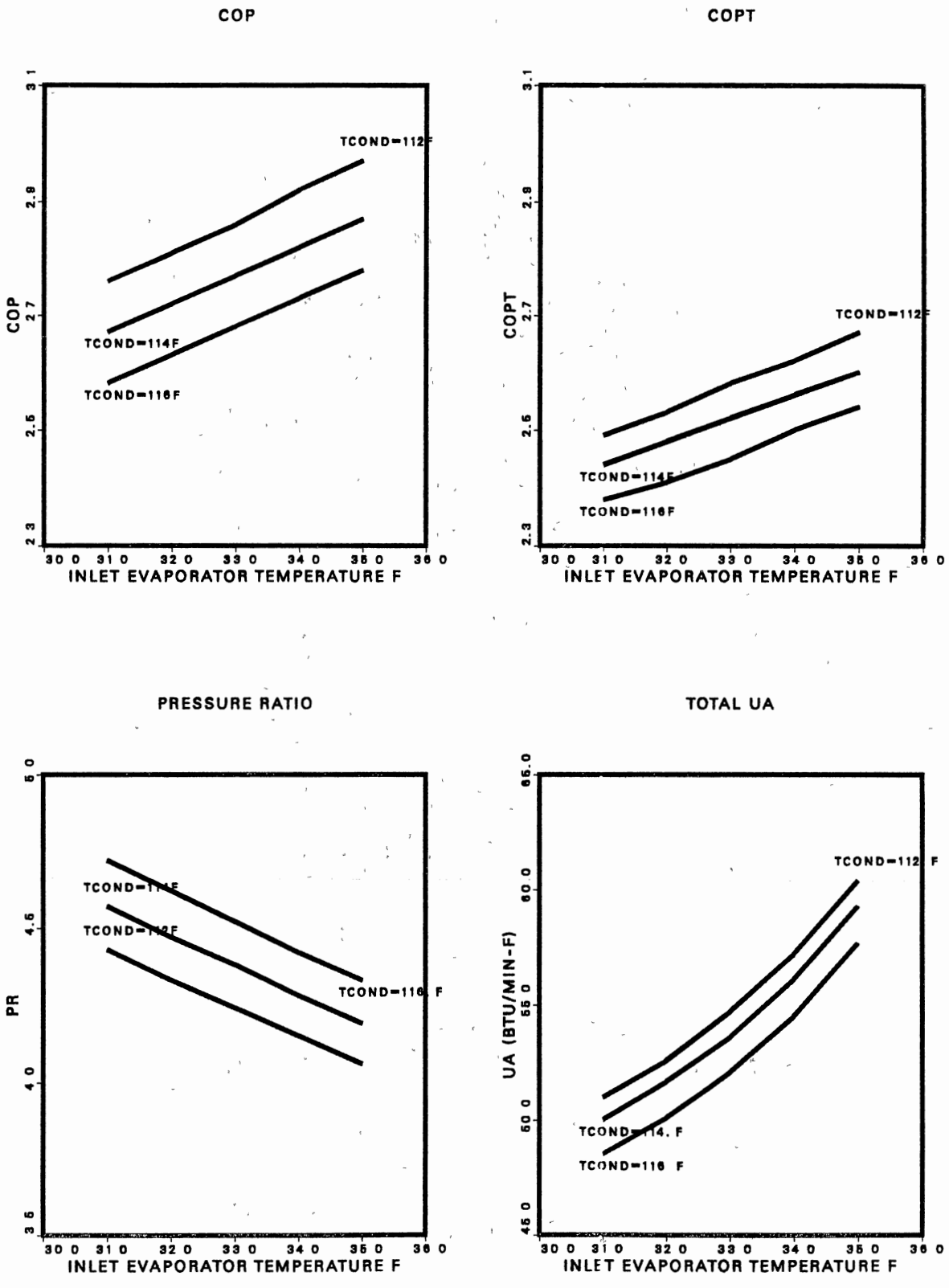
SENSITIVITY OF REFRIGERANT MIXTURE ON CYCLES



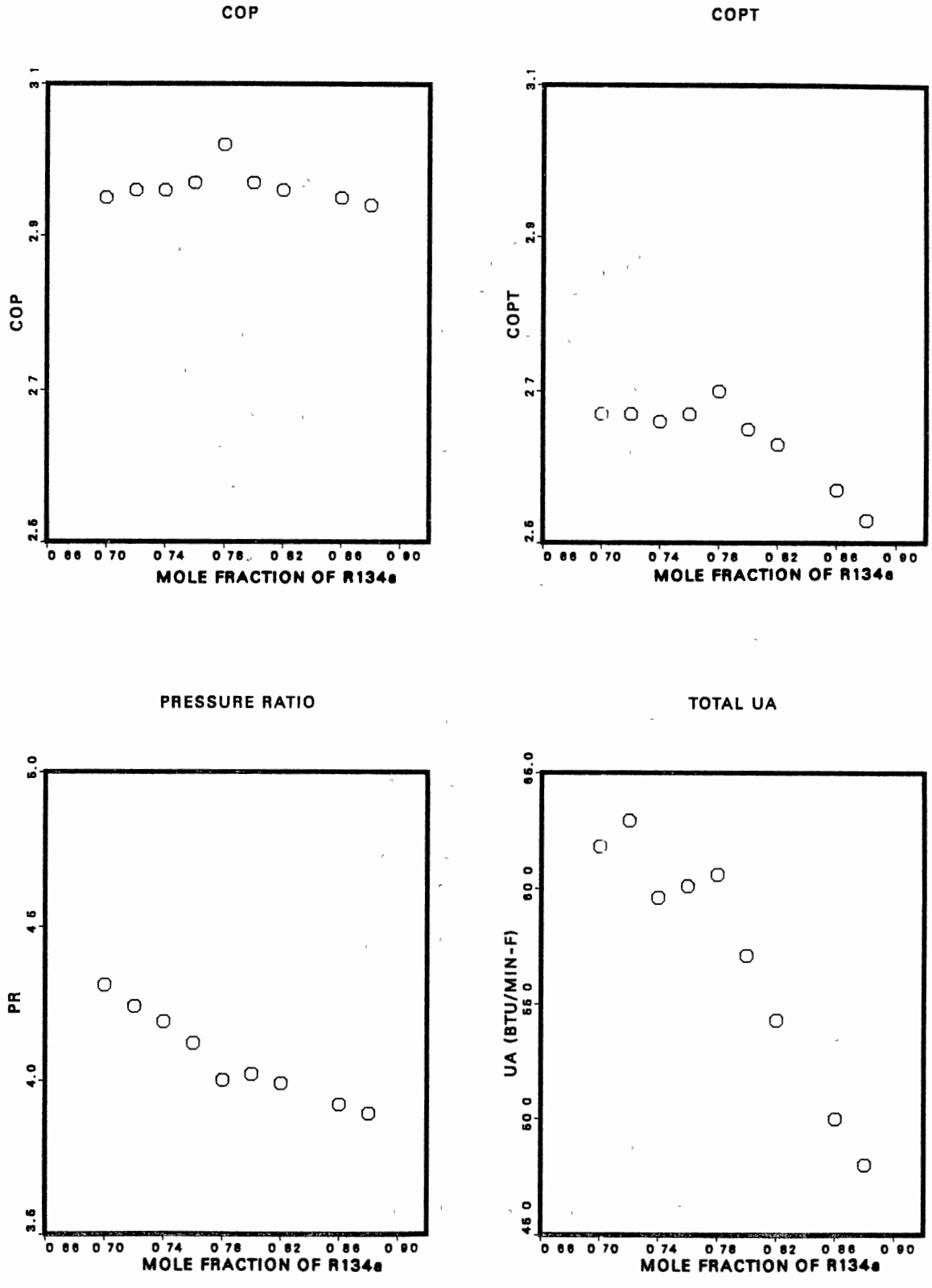
Sensitivity of R-134a on Cycle 1



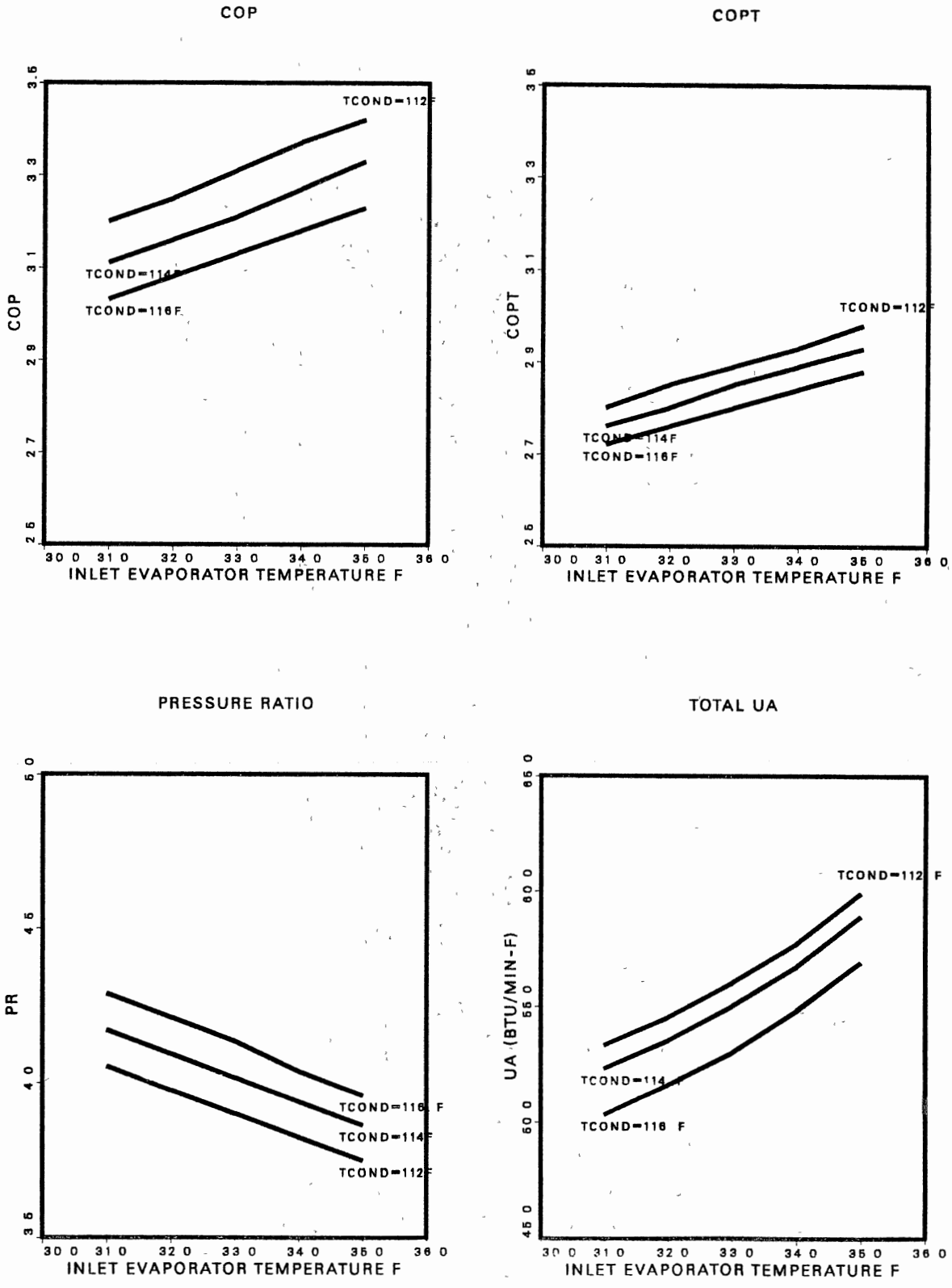
Sensitivity of R-134a on Cycle 2



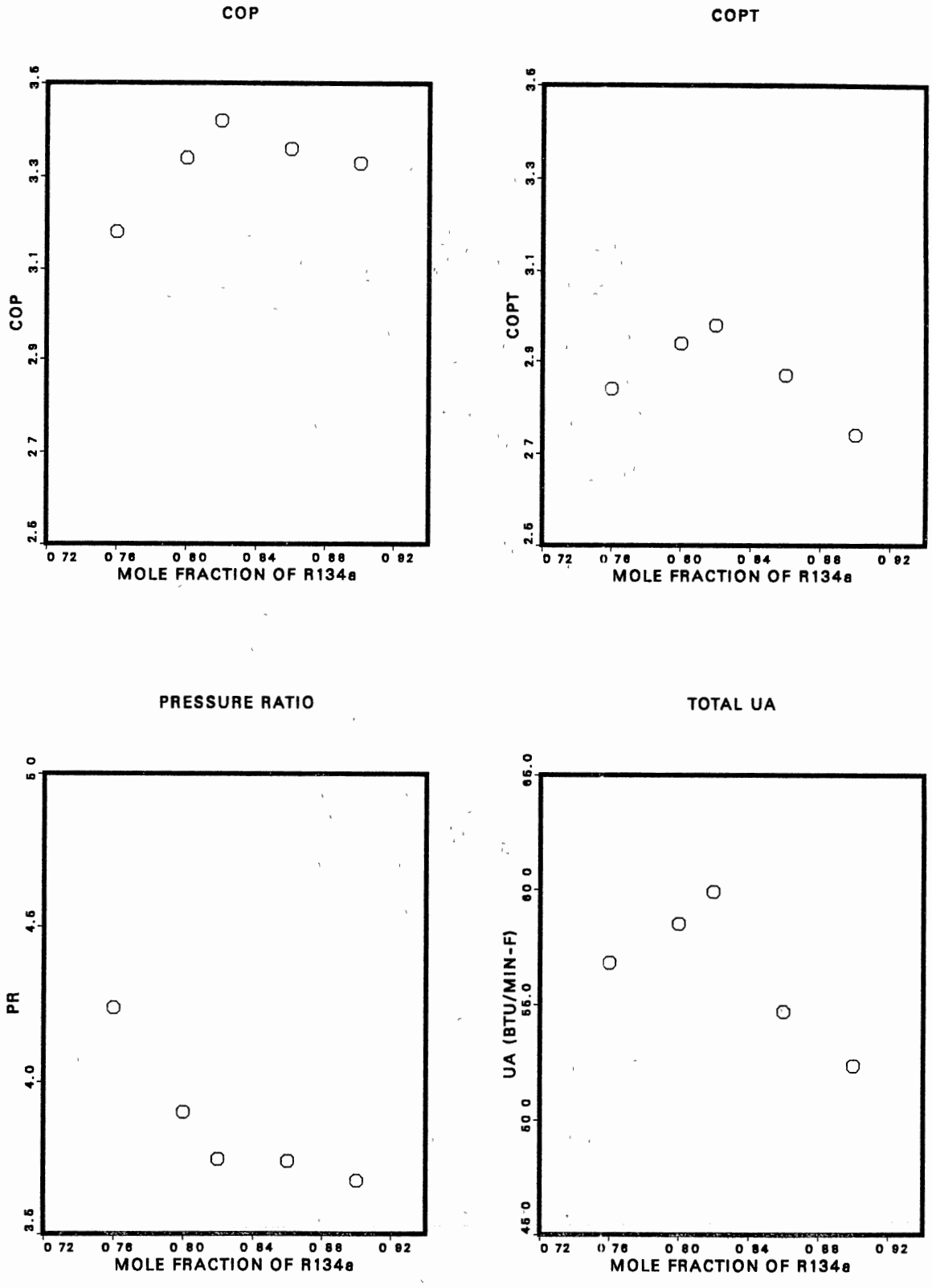
Sensitivity of 0.78 R-134a/0.22 R-123 on Cycle 3



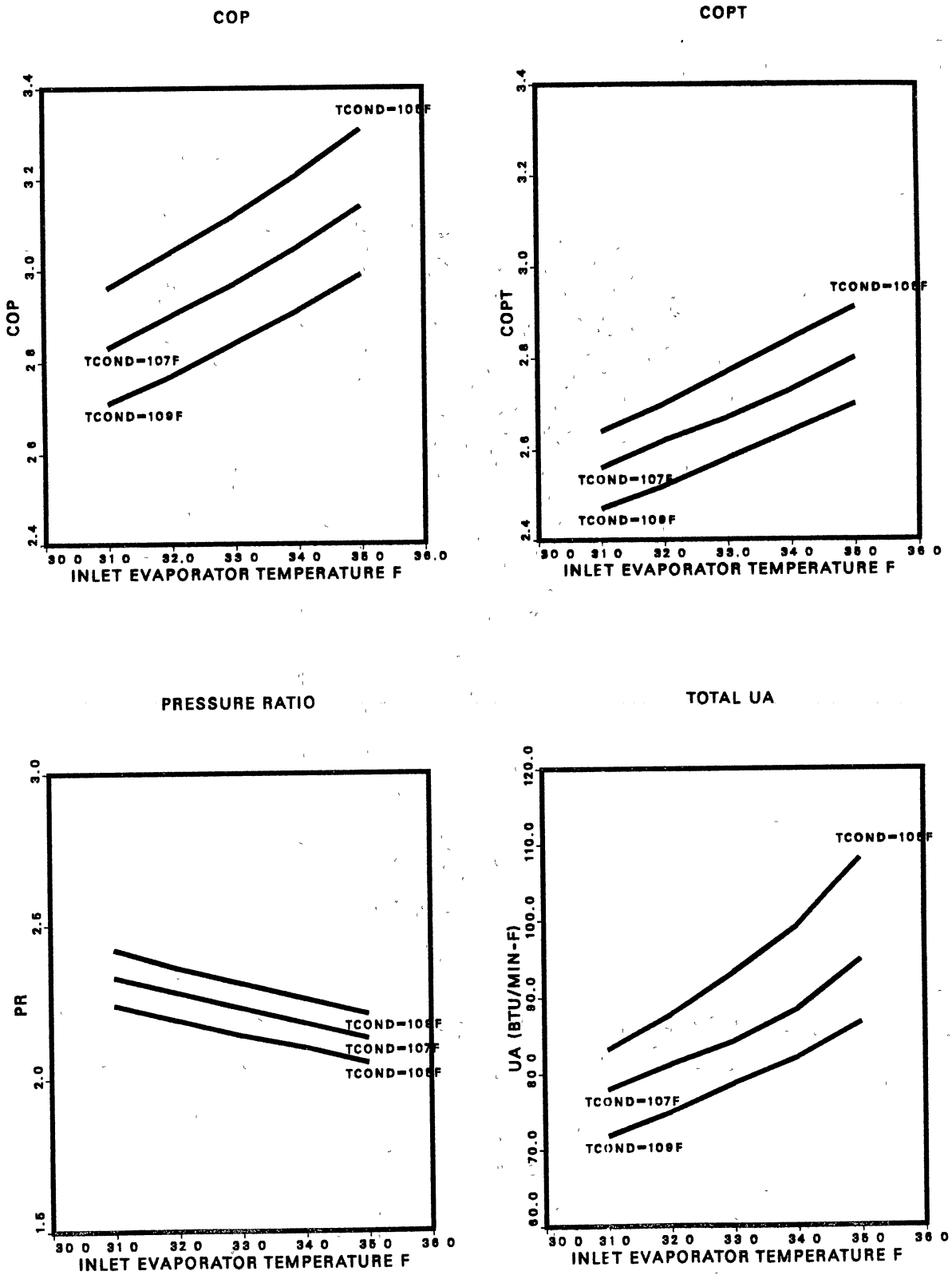
Sensitivity of R-134a/R-123 on Cycle 3



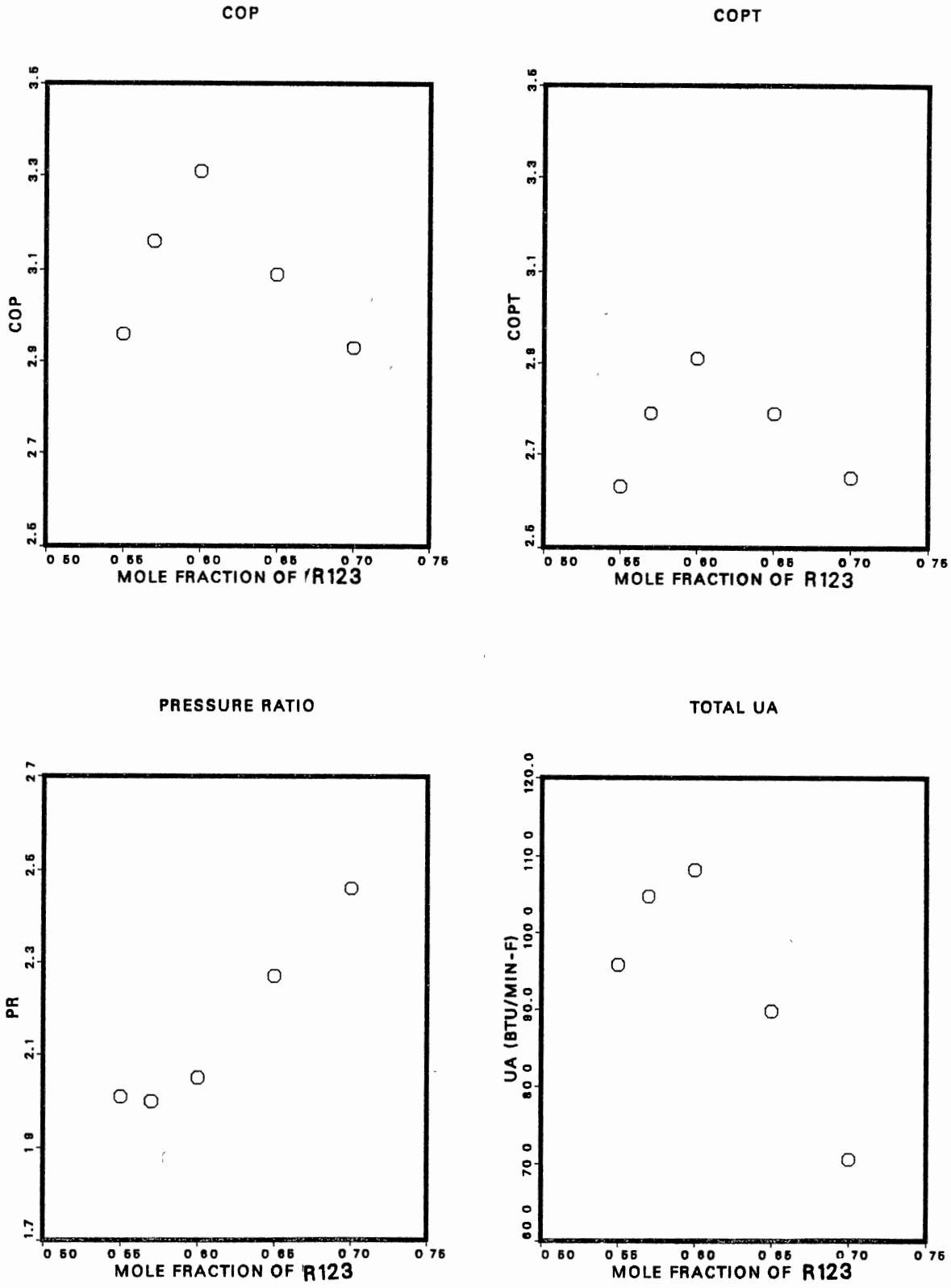
Sensitivity of 0.82 R-134a/0.18 R-123 on Cycle 4



Sensitivity of R134a/R123 on CYCLE 4



Sensitivity of 0.60 R-22/0.40 R-123 on Cycle 5



Sensitivity of R-22/R-123 on Cycle 5

APPENDIX D

PROGRAM LISTING

```
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C     MAIN PROGRAM READ INPUT DATA TO CREATE DO LOOP STRUCTURE
C     FOR CYCLES 1-6
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
      IMPLICIT REAL*8 (A-H,O-Z)
      CHARACTER*6 HREF(0:20),FLUD1,FLUD2,VCSN
      CHARACTER*20 TIT1,TIT2,TIT3,TIT4,TIT5,TIT6,TIT7,TIT8,TIT9,TIT10
      CHARACTER*20 TITC
      CHARACTER*17 TCOMP(50)
      LOGICAL LBUB,LCRIT
      COMMON /ESDATA/ COEFF(9,20),CRIT(5,20)
      COMMON /HREF1/ HREF,MMAX
      COMMON /HSZERO/ HZERO(20),SZERO(20)
      COMMON /TOLSH/ TOLH,TOLS
      COMMON /RDATA4/ R
      DIMENSION TT(20),PP(20),HH(20),SS(20),TLA(200),THA(200),DELT(200)
     #DIMENSION IVCS(6)
      DATA IVCS/6,0,6,1,5,2/
      DATA IFIRST/0/
      DATA TOLH,TOLS/0.00001,0.00001/
      OPEN(UNIT=5,FILE='THE3.DAT',STATUS='OLD')
      WRITE(6,*)' =1 PURE/CONVENTIONAL CYCLE'
      WRITE(6,*)' =2 PURE/2 STAGE COMPRESSOR CYCLE'
      WRITE(6,*)' =3 NARM/CONVENTIONAL CYCLE'
      WRITE(6,*)' =4 NARM/2 STAGE COMPRESSOR CYCLE'
      WRITE(6,*)' =5 NARM/TWO-PHASE OUTLET EVAPORATOR WITHOUT LIQUID PUM
      WRITE(6,*)' =6 NARM/TWO-PHASE OUTLET EVAPORATOR WITH LIQUID PUMP'
      WRITE(6,*) ' INPUT CYCLE NO '
      READ(5,*) ICYCLE
      ISH=IVCS(ICYCLE)
      WRITE(6,*) ' PRINT HX TEMPERATURE PROFILE 0  1'
      READ(5,*) IPRI
      WRITE(6,*) ' DRAW FLOW DIAGRAM  0 1'
      READ(5,*) IDIAG
      WRITE(6,*) ' CHANGE DO LOOP  0  1'
      READ(5,*) ILOOP
      IF (ILOOP.EQ.0) GOTO 89
      WRITE(6,*)'READ NUMBER OF DO LOOP FOR'
      WRITE(6,*)'EVAP TEMP,DEG SUPERHEAT,HX TEMP. DIFF,COND.SUB. DEGREE'
      READ (5,*) NUMT5,NUMSUP,NUMHX,NUMDSC
      WRITE(6,*)' INPUT INCREAMENT FOR'
      WRITE(6,*)'EVAP TEMP,DEG SUPERHEAT,HX TEMP. DIFF,COND.SUB. DEGREE'
      READ(5,*) T5DEL,DELSUP,TDELE,DELDSC
      CLOSE (UNIT=5)
      WRITE(6,*) 'NUMT5 NUMSUP NUMHX NUMDSC'
      WRITE(6,*) NUMT5, NUMSUP, NUMHX ,NUMDSC
 89   CONTINUE
C     UNIT 6 OUTPUT FILE
      OPEN (UNIT=06, STATUS='NEW', FILE= 'THE.OUT')
C     UNIT 90 SUMMARY OF DOLOOP FILE
      OPEN (UNIT=90, STATUS='NEW', FILE= 'VCS.SUM')
C     UNIT 75 FOR STATE POINT SCHEMATIC
      IF (IDIAG.EQ.1) OPEN (UNIT=75, STATUS='NEW', FILE= 'THECYCLE.INP')
C ISH=0 PURE , 2 STAGE COMPRESSOR
```

```
C ISH=1 NARM , 2 STAGE COMPRESSOR
C ISH=2 NARM , 2-PHASE OUTLET EVAPORATOR WITH LIQUID PUMP
C ISH=5 NARM , 2-PHASE OUTLET EVAPORATOR WITHOUT LIQUID PUMP
C ISH=6 PURE,NARM BASIC CYCLES
      IF (ISH.GT.0) GOTO 1
      OPEN (UNIT=80, STATUS='OLD', FILE= 'VCS0.INP')
      WRITE(90,*)' TCON TEVA DTHX DTSB COP  COPT PR1  PR2  FL41  FL31 UA
     *CO UAEV UAS2 UATO VSUC1 2'
      IF (IDIAG.EQ.1) WRITE(75,*)ISH,0,0,11
      GOTO 11
   1  IF (ISH.GT.1) GOTO 2
      OPEN (UNIT=80, STATUS='OLD', FILE= 'VCS1.INP')
      WRITE(90,*)' TCON TEVA DTHX COP  COPT PR1  PR2  FL41  FL31 UACO UA
     *EV UAS1 UAS2 UATO VSUC1 2'
      IF (IDIAG.EQ.1) WRITE(75,*)ISH,1,0,14
      GOTO 11
   2  IF (ISH.GT.2) GOTO 5
      OPEN (UNIT=80, STATUS='OLD', FILE= 'VCS2.INP')
      WRITE(90,*)' TCON TEVA DTSP COP  COPT PR1  FL31  FL41 UACO UAEV UA
     *S1 UAS2 UATO QUAE QUAMC'
      IF (IDIAG.EQ.1) WRITE(75,*)ISH,2,3,13
      GOTO 11
   5  IF (ISH.GT.5) GOTO 6
      OPEN (UNIT=80, STATUS='OLD', FILE= 'VCS5.INP')
      WRITE(90,*)' TCON TEVA DTSP COP  COPT PR1  FL41  FL31 UACO UAEV UA
     *S1 UAS2 UATO'
      IF (IDIAG.EQ.1) WRITE(75,*)ISH,2,0,9
      GOTO 11
   6  IF (ISH.GT.6) GOTO 7
      OPEN (UNIT=80, STATUS='OLD', FILE= 'VCS6.INP')
      WRITE(90,*)' TCON TEVA DTSB COP  COPT PR1  FL41  FL31 UACO UAEV UA
     *TO SUC V'
      IF (IDIAG.EQ.1) WRITE(75,*)ISH,0,-1,8
      GOTO 11
   7  CONTINUE
  11  CONTINUE
      IR = 80
      IW = 6
C     COMMOM VARIABLES FOR EVERY CYCLES
      READ (IR,*) FLUD1,FLUD2,VCSN
      READ (IR,10) TIT1,X,F0,F1
      READ (IR,20) TIT2,IU,NDIV,ICHECK,IFSI,IFSO
      READ (IR,10) TIT3,HEATL
      READ (IR,10) TIT4,T5I,DTD,DTSUPEI,DPEVA
      READ (IR,10) TIT5,DTHXI,DPHX1,DPHX2,DPHX
      READ (IR,10) TIT6,T4I,DPCO,DPCOOL,DTSUBI
      READ (IR,10) TIT7,PR1I,TIME,DPR
      READ (IR,10) TIT8,COMPEF,EFFMOT,DTSUP2
      READ (IR,10) TIT9,PUMPEF,FANEF,DPSI
      READ (IR,10) TIT10,T31,T32,T41,T42
      CLOSE (UNIT=80)
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C     VCSN      : CYCLE NUMBER
C     FLUD1     : REFRIGERANT 1 OF BINARY MIXTURE
C     FLUD2     : REFRIGERANT 2 OF BINARY MIXTURE
```

```
C     TIT1,...,TIT10 DUMMY VARIABLES USE FOR INPUT FILE TO REMIND THE
C                    USERS NAME OF VARIABLE
C     X          : MOLE FRACTION
C     F0,F1      : INTERACTION COEFFICIENTS
C     IU         : UNIT SYSTEM FOR OUTPUT   0 FOR SI AND 1 FOR ENGLISH
C     NDIV       : NUMBER OF SEGMENT TO ESTIMATE THE HEAT EXCHANGER
C                  CONDUCTANCE AND PINCH POINT TEMPERATURE
C     ICHECK     : TO CHECK HEAT EXCHANGERS (=0 : NO-CHECK)
C     IFSI,IFSO  : FLUID SINK AND SOURCE SIDE (=0 : AIR. =1: WATER)
C     HEATL      : EVAPORATOR LOAD (TON)
C     T5I        : INITIAL VALUE OF EVAPORATOR INLET TEMPERATURE (C)
C     DTD        : TEMPERATURE DROP ACROSS MAIN EXPANSION DEVICE (C)
C     DTSUPEI    : INITIAL DEGREE SUPERHEAT EVAPORATOR OUTLET (C)
C     DPEVA      : EVAPORATOR PRESSURE DROP (KPA)
C     DTHXI      : INITIAL DIFFERENTIAL TEMPERATURE OF HOT SIDE OUTLET
C                  AND COLD SIDE INLET OF SUBCOOLER 2 (C)
C     DPHX,1,2   : SUBCOOLER PRESSURE DROP FOR HOT AND COLD SIDE (C)
C
C     T4I        : INITIAL VALUE OF CONDENSER OUTLET TEMPERATURE (C)
C     DPCO       : CONDENSER PRESSURE DROP (KPA)
C     DPCOOL     : MOTOR COOLING PRESSURE DROP (KPA)
C     DTSUBI     : INITIAL VALUE OF DEGREE SUBCOOL OF CONDENSER (C)
C     PR1        : FIRST STAGE COMPRESSOR PRESSURE RATIO
C     TIME       : NUMBER OF DO LOOP FOR PRESSURE RATIO
C     DPR        : PRESSURE RATIO INCREASEMENT
C     COMPEF     : ISENTROPIC COMPRESSOR EFFICIENTCY
C     EFFMOT     : COMPRESSOR ELECTRIC MOTOR EFFICIENTCY
C     DTSUP2     : REQUIRED DEGREE SUPERHEAT AT INLET OF SECOND STAGE
C                  OF COMPRESSOR
C     PUMPEF     : PUMP EFFICIENTCY
C     FANEF      : CONDENSER FAN EFFICIENTCY
C     DPSI       : CONDENSER PRESSURE DROP SINK SIDE (KPA)
C     T31,T32    : INLET, OUTLET OF SOURCE SIDE TEMPERATURE (C)
C     T41,T42    : INLET, OUTLET OF SINK SIDE TEMPERATURE (C)
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

      WRITE(90,91) VCSN,X,FLUD1,1.-X,FLUD2
 91   FORMAT(5X,A6,2X,F4.2,1X,A6,'/',F4.2,1X,A6)
      WRITE (IW,*) FLUD1,FLUD2,VCSN
      WRITE (IW,10) TIT1,X,F0,F1
      WRITE (IW,20) TIT2,IU,NDIV,ICHECK,IFSI,IFSO
      WRITE (IW,10) TIT3,HEATL
      WRITE (IW,10) TIT4,T5I,DTD,DTSUPEI,DPEVA
      WRITE (IW,10) TIT5,DTHXI,DPHX1,DPHX2,DPHX
      WRITE (IW,10) TIT6,T4I,DPCO,DPCOOL,DTSUBI
      WRITE (IW,10) TIT7,PR1I,TIME,DPR
      WRITE (IW,10) TIT8,COMPEF,EFFMOT,DTSUP2
      WRITE (IW,10) TIT9,PUMPEF,FANEF,DPSI
      WRITE (IW,10) TIT10,T31,T32,T41,T42
      WRITE (IW,*)'*************** END OF INPUT FILE ***********'
10    FORMAT(A20,6F10.3)
20    FORMAT (A20,6I10)
C VCS0 ITERATION T5,DTHX,DTSUB
C VCS1 ITERATION T5,DTHX
C VCS2 ITERATION T5,DTSUPE
```

```
C VCS5 ITERATION T5,DTSUPE
C VCS6 ITERATION T5,DTSUB
C
  901 CONTINUE
C
C VCS0,VCS1,VCS2,VCS5,VCS6
C ITERATION EVAPORATING TEMPERATURE
      IF (ILOOP.EQ.0) NUMT5=3
C     T5DEL=1.
C      IF (IU.EQ.1) T5DEL=T5DEL/1.8
      DO 1000 NUM=1,NUMT5
      T5=T5I+(NUM-1)*T5DEL
      IF (ISH.EQ.6) GOTO 915
      IF (ISH.EQ.2.OR.ISH.EQ.3.OR.ISH.EQ.4.OR.ISH.EQ.5) GOTO 910
  905 CONTINUE
C
CVCS0,VCS1
C ITERATION LOW SUBCOOLER EFFECTIVENESS
C      TDELE=1.
C      IF (IU.EQ.1) TDELE=TDELE/1.8
      DTHX=DTHXI
      IF (ILOOP.EQ.0) NUMHX=3
      DO 990  INUM=1,NUMHX
      DTHX=DTHXI+TDELE*(INUM-1)
      WRITE(6,*)'     **** DTHX  =',DTHX
      IF (ISH.EQ.0) GOTO 915
      IF (ISH.EQ.1) GOTO 930
  910 CONTINUE
C
C VCS2,VCS3,VCS4,VCS5
C ITERATION SUPERHEATED EVAPORATOR OUT
C     DELSUP=1.
C     IF (IU.EQ.1) DELSUP=DELSUP/1.8
      IF (ILOOP.EQ.0) NUMSUP=3
      DO 980  ISUP=1,NUMSUP
      DTSUPE=DTSUPEI+(ISUP-1)*DELSUP
      IF (ISH.NE.6) GOTO 930
  915 CONTINUE
C
C VCS0,VCS6
C ITERATION CONDENSER OUT DEGREE SUBCOOLED
C     DELDSC=1./1.8
C     IF (IU.EQ.1) DELDSC=DELDSC/1.8
      IF (ILOOP.EQ.0) NUMDSC=3
      DO 970 IDSC=1,NUMDSC
      DTSUB=DTSUBI+(IDSC-1)*DELDSC
      WRITE(6,*)'            &&&&& DTSUB =',DTSUB
C
C
  930 CONTINUE
      IF (ISH.EQ.0) GOTO 50
      IF (ISH.EQ.1) GOTO 51
      IF (ISH.EQ.2) GOTO 52
      IF (ISH.EQ.5) GOTO 55
      IF (ISH.EQ.6) GOTO 56
 50   CALL VCS0(ISH,IPRI,IDIAG,IW,FLUD1,FLUD2,VCSN,X,F0,F1,
```

```
     *IU,NDIV,ICHECK,IFSI,IFSO,HEATL,T5,DTD,DTSUPEI,DPEVA,
     *DTHX,DPHX1,DPHX2,DPHX,T4,DPCO,DPCOOL,DTSUB,PR1I,TIME,DPR,
     *COMPEF,EFFMOT,DTSUP2,PUMPEF,FANEF,DPSI,T31,T32,T41,T42,COP)
      WRITE(6,*) ' DTHXI,DTHX,INUM',DTHXI,DTHX,INUM
      GOTO 100
 51   CALL VCS1(ISH,IPRI,IDIAG,IW,FLUD1,FLUD2,VCSN,X,FO,F1,
     *IU,NDIV,ICHECK,IFSI,IFSO,HEATL,T5,DTD,DTSUPE,DPEVA,
     *DTHX,DPHX1,DPHX2,DPHX,T4,DPCO,DPCOOL,DTSUB,PR1I,TIME,DPR,
     *COMPEF,EFFMOT,DTSUP2,PUMPEF,FANEF,DPSI,T31,T32,T41,T42,COP)
      GOTO 100
 52   CALL VCS2(ISH,IPRI,IDIAG,IW,FLUD1,FLUD2,VCSN,X,FO,F1,
     *IU,NDIV,ICHECK,IFSI,IFSO,HEATL,T5,DTD,DTSUPE,DPEVA,
     *DTHX,DPHX1,DPHX2,DPHX,T4,DPCO,DPCOOL,DTSUB,PR1I,TIME,DPR,
     *COMPEF,EFFMOT,DTSUP2,PUMPEF,FANEF,DPSI,T31,T32,T41,T42,COP)
      GOTO 100
 55   CALL VCS5(ISH,IPRI,IDIAG,IW,FLUD1,FLUD2,VCSN,X,FO,F1,
     *IU,NDIV,ICHECK,IFSI,IFSO,HEATL,T5,DTD,DTSUPE,DPEVA,
     *DTHXI,DPHX1,DPHX2,DPHX,T4,DPCO,DPCOOL,DTSUB,PR1I,TIME,DPR,
     *COMPEF,EFFMOT,DTSUP2,PUMPEF,FANEF,DPSI,T31,T32,T41,T42,COP)
      GOTO 100
 56   CALL VCS6(ISH,IPRI,IDIAG,IW,FLUD1,FLUD2,VCSN,X,FO,F1,
     *IU,NDIV,ICHECK,IFSI,IFSO,HEATL,T5,DTD,DTSUPE,DPEVA,
     *DTHX,DPHX1,DPHX2,DPHX,T4,DPCO,DPCOOL,DTSUB,PR1I,TIME,DPR,
     *COMPEF,EFFMOT,DTSUP2,PUMPEF,FANEF,DPSI,T31,T32,T41,T42,COP)
      GOTO 100
 100  CONTINUE
 970  CONTINUE
 980  CONTINUE
 990  CONTINUE
 1000 CONTINUE
      IF (IDIAG.EQ.1) CLOSE (UNIT=75)
      CLOSE (UNIT=6)
      STOP
      END
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C                CYCLE 1/PURE   CONVENTIONAL CYCLE
C                CYCLE 3/NARM   CONVENTIONAL CYCLE
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
      SUBROUTINE VCS6(ISH,IPRI,IDIAG,IW,FLUD1,FLUD2,VCSN,X,FO,F1,
     *IU,NDIV,ICHECK,IFSI,IFSO,HEATL,T5,DTD,DTSUPE,DPEVA,
     *DTHX,DPHX1,DPHX2,DPHX,T4,DPCO,DPCOOL,DTSUB,PR1I,TIME,DPR,
     *COMPEF,EFFMOT,DTSUP2,PUMPEF,FANEF,DPSI,T31,T32,T41,T42,COP)
      IMPLICIT REAL*8 (A-H,O-Z)
      CHARACTER*6 HREF(0:20),FLUD1,FLUD2,VCSN
      CHARACTER*20 TITC
      CHARACTER*17 TCOMP(50)
      LOGICAL LBUB,LCRIT
      COMMON /ESDATA/ COEFF(9,20),CRIT(5,20)
      COMMON /HREF1/ HREF,MMAX
      COMMON /HSZERO/ HZERO(20),SZERO(20)
      COMMON /TOLSH/ TOLH,TOLS
      COMMON /RDATA4/ R
      DIMENSION TT(20),PP(20),HH(20),SS(20),TLA(200),THA(200),DELT(200)
```

```
      DIMENSION CPP(2)
      DATA CPP/1.0,4.18/
      DATA IFIRST/0/
      DATA TOLH,TOLS/0.00001,0.00001/
C
C                    ****** NOTES *****
C
C  THE QUALITY CALCULATED BY THE NBS ROUTINES IS A MOLAR QUALITY
C
C
C
C  INITIALIZE COMMON BLOCKS FOR PURE COMPONENTS
C
      DO 100 I = 1,20
         IF (FLUD1 .EQ. HREF(I)) IR1 = I
         IF (FLUD2 .EQ. HREF(I)) IR2 = I
 100  CONTINUE
C
      CALL BCONST(IR1,IR2,F0,F1)
      WM1=CRIT(1,IR1)
      WM2=CRIT(1,IR2)
      XMOL=X
      WM=XMOL*WM1+(1.0-XMOL)*WM2
C     TIRR DEAD ZONE TEMPERATURE AS A BASE TEMPERATURE TO CALCULATE
C     IRREVERSIBILITY
      TIRR=298.
C
C (*3) SATURATED
      T3=T4+DTSUB
      LBUB = .TRUE.
      CALL BUBLT(T3,XMOL,XV3,P3,VL3,VV3,LBUB,LCRIT)
      P4=P3
      LBUB = .TRUE.
      CALL BUBLT(T4,XMOL,XV4,P4S,VL4,VV4,LBUB,LCRIT)
      CALL HCVCPS(1,T4,VL4,XMOL,H4,CP,CP,VS)
      CALL HPIN(H4,P4,XMOL,T4TAM,YQ4,XL4,XV4,VL4,VV4,HL4,HV4)
      WRITE(6,*)' H4,P4,T4TAM',H4,P4,T4TAM
C    ENTROPY OUTLET OF CONDENSER
      SL4=ENTROP(T4,VL4,XL4)
      SS4=SL4
      H5=H4
C
C  AT H5 AND T5, OBTAIN P5,YQ5,VL5,ETC, FOR TWO PHASE MIXTURE
C
      IF (XMOL.LE.0.001.OR.XMOL.GE.0.999) THEN
      CALL BUBLT(T5,XMOL,XV5,P5S,VL5,VV5,.TRUE.,LCRIT)
      CALL HCVCPS(1,T5,VL5,XMOL,HL5,CP,CP,VS)
      CALL HCVCPS(1,T5,VV5,XMOL,HV5,CP,CP,VS)
      YQ5=(H5-HL5)/(HV5-HL5)
      P5=P5S
      XL5=XMOL
      XV5=XMOL
      ELSE
      CALL HTIN(H5,T5,XMOL,P5,YQ5,XL5,XV5,VL5,VV5,HL5,HV5)
      ENDIF
C   ENTROPY INLET TO EVAPORATOR
```

```
      SL5=ENTROP(T5,VL5,XL5)
      SV5=ENTROP(T5,VV5,XV5)
      S5=YQ5*SV5+(1.-YQ5)*SL5
      SS5=S5
C
C  AT P1  OBTAIN T1,YQ1,VL1,ETC.
C
      P1=P5-DPEVA
      CALL BUBLP(P1,XML,XMOL,T1V,VL1,VV1,.FALSE.,LCRIT)
      T1=T1V+DTSUPE
      IF (XMOL.LE.0.001.OR.XMOL.GE.0.999) THEN
      CALL HCVCPS(1,T1,VV1,XMOL,H1,CP,CP,VS)
      CALL HTIN(H1,T1,XMOL,P1PRI,YQ1,XL1,XV1,VL1,VV1,HL1,HV1)
      ELSE
      HMIN = H5
      HMAX = HV5*2.
      CALL PTIN(1,HMIN,HMAX,P1,T1,XMOL,H1,YQ1,XL1,XV1,
     1              VL1,VV1,HL1,HV1)
      ENDIF
       SL1 = ENTROP(T1,VL1,XL1)
       SV1 = ENTROP(T1,VV1,XV1)
      VVCOM=VV1
      IF (YQ1.LE.0.) S1=SL1
      IF (YQ1.GT.0..AND.YQ1.LT.1.) S1= YQ1*SV1 + (1.0-YQ1)*SL1
      IF (YQ1.GT.1.0)S1=SV1
      SS1=S1
      T21=T1
      P21=P1
      S21=S1
      SS21=S21
      H21=H1
      YQ21=YQ1
      DM1=0
C
C
C     COMPRESSOR
      ITIME=TIME
C     DO 130 IPR=1,ITIME
C     PR1=PR1I+DPR*(IPR-1)
      PR1MI=1.001
      PR1MA=2.*PR1I
      IIPR=0
      PR1=PR1I
 141  CONTINUE
      P22=P21*PR1
      CALL SPIN(S21,P22,XMOL,T22S,YQ22,XL22,XV22,VL22,VV22,
     *SL22,SV22)
      CALL HCVCPS(1,T22S,VV22,XV22,H22S,CV,CP,VS)
      H22 = (H22S-H21)/COMPEF + H21
      CALL HPIN(H22,P22,XMOL,T22,YQ22,XL22,XV22,VL22,VV22,
     *HL22,HV22)
      SS22=ENTROP(T22,VV22,XV22)
C ITERATION FOR FLOW RATE MOTOR COOLING
      P20=P21
      H20=H21
      T20=T21
```

```
      SS20=SS21
      P19=P20+DPCOOL
C
      DM2=0.1
      DM2MA=10.
      DM2MI=0.00001
      IDM=0
 140  CONTINUE
      DIFH=(1.+DM2)*(H22-H21)
      ELECP=DIFH/EFFMOT
      COOL=ELECP-DIFH
      H19P=H20-COOL/DM2
      IF (ABS(H19P/H4-1.).LE.0.001) GOTO 160
      CALL CNVRG(DM2,DM2MI,DM2MA,H4,H19P,IDM,DUM3,DUM4,6)
      IF (IDM-30) 140,140,150
 150  STOP
 160  H19=H19P
      IF (XMOL.LE.0.001.OR.XMOL.GE.0.999) THEN
      CALL BUBLP(P19,XMOL,XML,T19,VL19,VV19,.TRUE.,LCRIT)
      CALL HCVCPS(1,T19,VL19,XMOL,HL19,CV,CP,VS)
      CALL HCVCPS(1,T19,VV19,XMOL,HV19,CV,CP,VS)
      YQ19=(H19-HL19)/(HV19-HL19)
      XL19=XMOL
      XV19=XMOL
      ELSE
      CALL HPIN(H19,P19,XMOL,T19,YQ19,XL19,XV19,VL19,VV19
     1,HL19,HV19)
      ENDIF
C(19) ENTROPY   M/C INLET
       SL19 = ENTROP(T19,VL19,XL19)
       SV19 = ENTROP(T19,VV19,XV19)
      IF (YQ19.LE.0.) S19=SL19
      IF (YQ19.GT.0..AND.YQ19.LT.1.) S19= YQ19*SV19+(1.0-YQ19)*SL19
      IF (YQ19.GT.1.0)S19=SV19
      SS19=S19
      P4=P22-DPCO
      FCHECK= P4/P3
      IF (ABS(FCHECK-1.).LE.0.001) GO TO 191
      CALL CNVRG(PR1,PR1MI,PR1MA,FCHECK,1.,IIPR,DUM10,DUM11,6)
      IF (IIPR-30) 141,141,192
 192  STOP
 191  CONTINUE
      CALL HPIN(H4,P4,XMOL,T44,YQ4,XL44,XV44,VL44,VV44
     1,HL44,HV44)
      T20=T21
      H20=H21
C (7) SPLIT FROM (4)
      FL1=1.
      T7=T4
      P7=P4
      H7=H4
      FL7=FL1
      SS7=SS4
      SS6=SS4
      YQ7=YQ4
      FL5=FL1
```

```
      FL21=FL1+DM2
      FL22=FL21
      FL4=FL22
      FL19=DM2
      FL20=FL19
C     M/C QUALITY OUT
      YQ20=YQ21
C CALCULATE FOR DESIGN HEAT LOAD HEATL(TONS)
C 1 TON = 3.52 KJ/SEC = 211.2 KJ/MIN
C EVMOL  EVAP FLOWRATE MOLES/MIN
      QLOAE= HEATL*211.2
      EVMOL=QLOAE/(H1-H5)
      VRATE=FL21*VVCOM*EVMOL
      EE=EVMOL
      IF (IPRI.EQ.0) GOTO 770
C     PRINTOUT THE OUTPUT DATA
      CALL CONAME(IU,ISH,TCOMP)
      CALL UNIT(IU,TCOMP, 1,T1,P1,H1,FL1,YQ1,WM,EE,XMOL,SS1)
      CALL UNIT(IU,TCOMP, 5,T5,P5,H5,FL5,YQ5,WM,EE,XMOL,SS5)
      CALL UNIT(IU,TCOMP, 4,T4,P4,H4,FL4,YQ4,WM,EE,XMOL,SS4)
      CALL UNIT(IU,TCOMP, 7,T7,P7,H7,FL7,YQ7,WM,EE,XMOL,SS7)
      CALL UNIT(IU,TCOMP,20,T20,P20,H20,FL20,YQ20,WM,EE,XMOL,SS20)
      CALL UNIT(IU,TCOMP,21,T21,P21,H21,FL21,YQ21,WM,EE,XMOL,SS21)
      CALL UNIT(IU,TCOMP,22,T22,P22,H22,FL22,YQ22,WM,EE,XMOL,SS22)
      CALL UNIT(IU,TCOMP,19,T19,P19,H19,FL19,YQ19,WM,EE,XMOL,SS19)
 770  CONTINUE
C     SINK OUTLET TEMPERATURE  (5 F PINCH POINT)
      DTPIN=5./1.8
      IF (YQ22.LE.1) THEN
      T42=T22-DTPIN
      ELSE
      CALL BUBLP(P22,XXX,XV22,T22VS,VL22S,VV22S,.FALSE.,LCRIT)
      CALL HCVCPS(1,T22VS,VV22S,XV22,H22S,CV,CP,VS)
      RAT=(H22-H4)/(H22S-H4)
      T42=(T22VS-DTPIN-T41)*RAT+T41
      ENDIF
      CALL TES(IU,1+DM1+DM2,T41,T22,T4,T42,0.,H22,H4,0.,0.,P22,P4,0.,
     *XMOL,0.,NDIV,EVMOL,UASIC,-1,3)
      Q41=FL22*(H22-H4)
      FL41=Q41/(T42-T41)/CPP(IFSI)
      Q31=(H1-H5)
      FL31=Q31/(T31-T32)/CPP(IFSO)
      CALL SINSOU(FEOUT,FCOUT,IU,FL31,T31,T32,FL41,T41,T42,EE)
      QEVA1=(H1-H5)/WM
      COP= (H1-H5)/ELECP
      IF (IFSI.EQ.1) RHO=101.325/0.287/T41
      POFAN=FL41*DPSI/RHO/FANEF
      ELECT=POFAN+ELECP
      COPT=(H1-H5)/ELECT
      POPUMP=POFAN/125.
      ELECTP=POPUMP+ELECP
      COPP=(H1-H5)/ELECTP
      IF (IU.EQ.0) WRITE(6,125)T5,DTSUB,QEVA1,VRATE
      IF (IU.EQ.1) WRITE(6,126)T5*1.8-460.,DTSUB*1.8,.948*QEVA1/2.20462
     *,VRATE*264.15
      WRITE(6,127)COP,COPT,COPP,PR1,YQ20
```

```
 125  FORMAT(//,30X,'  TEM EVAP IN (K)        =',F10.1,
     &/,         30X,'  DT SUBCOOLED (K)       =',F10.1,
     &/,         30X,'  COOLING RATE (KJ/KG)   =',F10.1,
     &/,         30X,'  SUC VOL RATE(M**3/MIN) =',F10.1)
 126  FORMAT(//,30X,'  TEM EVAP IN (F)        =',F10.1,
     &/,         30X,'  DT SUBCOOLED (F)       =',F10.1,
     &/,         30X,'  COOLING RATE (BTU/LBM) =',F10.1,
     &/,         30X,'  SUC VOL RATE (GPM)     =',F10.1)
 127  FORMAT(
     &           30X,'  COP                    =',F10.2,
     &/,         30X,'  COPT                   =',F10.2,
     &/,         30X,'  COPP                   =',F10.2,
     &/,         30X,'  PRESSURE RATIO         =',F10.2,
     &/,         30X,'  QUALITY M/C OUT        =',F10.2,//)
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C IRREVERSIBILITY ANALYSIS
      IRRE=1
      IF (IRRE.EQ.1) THEN
      WRITE(6,*)'     IRREVERSIBILITY ANALYSIS '
      IF (IU.EQ.0) WRITE(6,700)TIRR
      IF (IU.EQ.1) WRITE(6,701)TIRR*1.8-460.
  700 FORMAT(6X,' AT INLET SINK TEMPERATURE (K) =',F8.1,/,
     &5X,'FLOW RATE      ENTROPY           IRRE.',/,
     &5X,' KG/MIN         KJ/KG-K              ',/)
  701 FORMAT(6X,' AT INLET SINK TEMPERATURE (F) =',F8.1,/,
     &5X,'FLOW RATE      ENTROPY           IRRE.',/,
     &5X,' LBM/MIN      BTU/LBM-R              ',/)
C VALVE 7-5
      CALL FIRRE2(IU,'VALVE 7-5      ',TIRR,FL7,SS7,SS5,WM,EE,0.,0.
     $,TG1)
C VALVE 6-19
      CALL FIRRE2(IU,'VALVE 6-19     ',TIRR,FL19,SS6,SS19,WM,EE,0.,0.
     $,TG2)
C M/C 19-20
      CALL FIRREC(IU,'M/C 19-20      ',TIRR,.5*(T21+T22),FL19,H19,H20,
     $SS19,SS20,WM,EE,0.,0.,TG3)
C COMP 21-22
      CALL FIRRE2(IU,'COMP 21-22     ',TIRR,FL21,SS21,SS22,WM,EE,0.,
     $0.,TG4)
C COND 22-4 & 41-42
      CALL FIRRE4(IU,'COND 22-4,41-42',TIRR,FL22,H22,H4,SS22,SS4,WM,
     *FL41,T41,T42,1.,EE,IFSI,TG5)
C EVAP 5-1 & 31-32
      CALL FIRRE4(IU,'EVAP 5-1,31-32 ',TIRR,FL5,H5,H1,SS5,SS1,WM,
     *FL31,T31,T32,1.,EE,IFSO,TG6)
      TGTOT=TG1+TG2+TG3+TG4+TG5+TG6
C
      WRITE(6,447) TGTOT
 447  FORMAT('*************** TOTAL = ',F10.2,//)
      ENDIF
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
      IF (ICHECK.EQ.0) GOTO 130
      IF (IPRI.EQ.1) WRITE(6,*)' CONDENSER  3'
       CALL TES(IU,1+DM1+DM2,T41,T22,T4,T42,0.,H22,H4,0.,0.,P22,P4,0.,
```

```
     *XMOL,0.,NDIV,EVMOL,UASIC,IPRI,3)
      IF (IPRI.EQ.1) WRITE(6,*) ' EVAPORATOR 4'
      CALL TES(IU,0.,T5,T31,T32,T1,H5,0.,0.,H1,P5,0.,0.,P1,0.,
     *XMOL,NDIV,EVMOL,UASIE,IPRI,4)
      UATOTT=UASIC+UASIE
      IF (IU.EQ.0) WRITE(90,925)T4,T5,DTSUB,COP,COPT,PR1,FCOUT,FEOUT,
     *UASIC,UASIE,UATOTT,VRATE
      IF (IU.EQ.1)  WRITE(90,925)T4*1.8-460.,T5*1.8-460.,DTSUB*1.8,COP,
     *COPT,PR1,FCOUT,FEOUT,UASIC*.5267,UASIE*.5267,UATOTT*.5267,
     *VRATE*264.15
 925  FORMAT(1X,3F5.1,3F5.2,F6.1,4F5.1,F5.0)
 130  CONTINUE
      RETURN
      END
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C                     CYCLE 5 /NARM
C              TWO PHASE OUTLET AT EVAPORATOR WITHOUT LIQUID PUMP
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
      SUBROUTINE VCS5(ISH,IPRI,IDIAG,IW,FLUD1,FLUD2,VCSN,X,FO,F1,
     *IU,NDIV,ICHECK,IFSI,IFSO,HEATL,T5,DTD,DTSUPE,DPEVA,
     *DTHX,DPHX1,DPHX2,DPHX,T4,DPCO,DPCOOL,DTSUB,PR1I,TIME,DPR,
     *COMPEF,EFFMOT,DTSUP2,PUMPEF,FANEF,DPSI,T31,T32,T41,T42,COP)
      IMPLICIT REAL*8 (A-H,O-Z)
      CHARACTER*6 HREF(0:20),FLUD1,FLUD2,VCSN
      CHARACTER*20 TITC
      CHARACTER*17 TCOMP(50)
      LOGICAL LBUB,LCRIT
      COMMON /ESDATA/ COEFF(9,20),CRIT(5,20)
      COMMON /HREF1/ HREF,MMAX
      COMMON /HSZERO/ HZERO(20),SZERO(20)
      COMMON /TOLSH/ TOLH,TOLS
      COMMON /RDATA4/ R
      DIMENSION TT(20),PP(20),HH(20),SS(20),TLA(200),THA(200),DELT(200)
      DIMENSION CPP(2)
      DATA CPP/1.0,4.18/
      DATA IFIRST/0/
      DATA TOLH,TOLS/0.00001,0.00001/
C
C                     ****** NOTES *****
C
C  THE QUALITY CALCULATED BY THE NBS ROUTINES IS A MOLAR QUALITY
C
C
C
C  INITIALIZE COMMON BLOCKS FOR PURE COMPONENTS
C
      DO 100 I = 1,20
         IF (FLUD1 .EQ. HREF(I)) IR1 = I
         IF (FLUD2 .EQ. HREF(I)) IR2 = I
 100  CONTINUE
C
      CALL BCONST(IR1,IR2,FO,F1)
```

```
      WM1=CRIT(1,IR1)
      WM2=CRIT(1,IR2)
      XMOL=X
      WM=XMOL*WM1+(1.0-XMOL)*WM2
C     TIRR DEAD ZONE TEMOERATURE AS A BASE TEMPERATURE TO CALCULATE
C     IRREVERSIBILITY
      TIRR=298.
C
      T6= T5 + DTD
      LBUB = .TRUE.
C
C  AT T6 AND SATURATED LIQUID CONDITION, OBTAIN H6
C
      CALL BUBLT(T6,XMOL,XV6,P6S,VL6S,VV6S,LBUB,LCRIT)
      CALL HCVCPS(1,T6,VL6S,XMOL,H6,CP,CP,VS)
      H5 = H6
C
C  AT H5 AND T5, OBTAIN P5,YQ5,VL5,ETC, FOR TWO PHASE MIXTURE
C
      CALL HTIN(H5,T5,XMOL,P5,YQ5,XL5,XV5,VL5,VV5,HL5,HV5)
C  ENTROPY INLET TO EVAPORATOR
      SL5=ENTROP(T5,VL5,XL5)
      SV5=ENTROP(T5,VV5,XV5)
      S5=YQ5*SV5+(1.-YQ5)*SL5
      SS5=S5
C
C  AT P1  OBTAIN T1,YQ1,VL1,ETC.
C
      P1=P5-DPEVA
      CALL BUBLP(P1,XML,XMOL,T1V,VL1,VV1,.FALSE.,LCRIT)
      T1=T1V+DTSUPE
      HMIN = H5
      HMAX = HV5*1.5
      CALL PTIN(1,HMIN,HMAX,P1,T1,XMOL,H1,YQ1,XL1,XV1,
     1              VL1,VV1,HL1,HV1)
       SL1 = ENTROP(T1,VL1,XL1)
       SV1 = ENTROP(T1,VV1,XV1)
      IF (YQ1.LE.0.) S1=SL1
      IF (YQ1.GT.0..AND.YQ1.LT.1.) S1= YQ1*SV1 + (1.0-YQ1)*SL1
      IF (YQ1.GT.1.0)S1=SV1
C (1) EVAPORATOR OUT
      SS1=S1
      P8=P1-DPHX1
      P21=P8-DPCOOL
      ITIME=TIME
      DO 130 IPR=1,ITIME
      PR1=PR1I+DPR*(IPR-1)
      P22=PR1*P21
      P4=P22-DPCO
C
C  AT T4 AND P4 CONDITION OBTAIN H4,ETC
C
      HMIN=H5
      HMAX=3*H1
      CALL PTIN(1,HMIN,HMAX,P4,T4,XMOL,H4,YQ4,XL4,XV4,
     1              VL4,VV4,HL4,HV4)
```

```
C (4) ENTROPY CONDENSER OUTLET
       SL4 = ENTROP(T4,VL4,XL4)
       SV4 = ENTROP(T4,VV4,XV4)
      IF (YQ4.LE.0.) S4=SL4
      IF (YQ4.GT.0..AND.YQ4.LT.1.) S4= YQ4*SV4 + (1.0-YQ4)*SL4
      IF (YQ4.GT.1.0)S4=SV4
      SS4=S4
C   PRESSURE AT STATE POINT 4,7,6
      P7=P4-DPHX1
      P6=P7-DPHX2
      CALL HPIN(H6,P6,XMOL,T6,YQ6,XL6,XV6,VL6,VV6,HL6,HV6)
C (6) ENTROPY SUBLOW HOT OUTLET
       SL6 = ENTROP(T6,VL6,XL6)
       SV6 = ENTROP(T6,VV6,XV6)
      IF (YQ6.LE.0.) S6=SL6
      IF (YQ6.GT.0..AND.YQ6.LT.1.) S6=YQ6*SV6 +(1.0-YQ6)*SL6
      IF (YQ6.GE.1.0) S6=SV6
      SS6=S6
C START ITERATION TO FIND T7
      XDT7=DTHX
      XDT7MI=0.
      XDT7MA=40.
      IDT7=0
 456  T7=T1+XDT7
      HMIN = H5/2.
      HMAX = HV5*2.
      CALL PTIN(1,HMIN,HMAX,P7,T7,XMOL,H7,YQ7,XL7,XV7,
     1              VL7,VV7,HL7,HV7)
C (7) ENTROPY SUB LOW HOT INLET
       SL7 = ENTROP(T7,VL7,XL7)
       SV7 = ENTROP(T7,VV7,XV7)
      IF (YQ7.LE.0.) S7=SL7
      IF (YQ7.GT.0..AND.YQ7.LT.1.) S7=YQ7*SV7 +(1.0-YQ7)*SL7
      IF (YQ7.GE.1.0)S7=SV7
      SS7=S7
C    CALCULATE FLOWRATE FOR SUB II BASE 1 MOLE OF EVAP FLOWRATE)
      DM1=(H5-H7)/(H7-H1)
      H8=H4-H7+H1
      CALL HPIN(H8,P8,XMOL,T8,YQ8,XL8,XV8,VL8,VV8,HL8,HV8)
C (8) M/C IN SUB HIGH COLD OUT
       SL8 = ENTROP(T8,VL8,XL8)
       SV8 = ENTROP(T8,VV8,XV8)
      IF (YQ8.LE.0.) S8=SL8
      IF (YQ8.GT.0..AND.YQ8.LT.1.)S8= YQ8*SV8 + (1.0-YQ8)*SL8
      IF (YQ8.GT.1.0)S8=SV8
      SS8=S8
      CALL BUBLP(P21,XML,XMOL,T21V,VL21,VV21,.FALSE.,LCRIT)
      T21=T21V+DTSUP2
      HMIN =HL8
      HMAX =HV8*2.0
      CALL PTIN(1,HMIN,HMAX,P21,T21,XMOL,H21,YQ21,XL21,XV21,
     1              VL21,VV21,HL21,HV21)
C (21) M/C OUT, COMPRESSOR IN
       SL21 = ENTROP(T21,VL21,XL21)
       SV21 = ENTROP(T21,VV21,XV21)
      VVCOM=VV21
```

```
      IF (YQ21.LE.0.) S21=SL21
      IF (YQ21.GT.0..AND.YQ21.LT.1.)S21= YQ21*SV21 + (1.0-YQ21)*SL21
      IF (YQ21.GT.1.0)S21=SV21
      SS21=S21
      CALL SPIN(S21,P22,XMOL,T22S,YQ22,XL22,XV22,VL22,VV22,
     1SL22,SV22)
      CALL HCVCPS(1,T22S,VV22,XV22,H22S,CV,CP,VS)
      H22 = (H22S-H21)/COMPEF + H21
      CALL HPIN(H22,P22,XMOL,T22,YQ22,XL22,XV22,VL22,VV22
     1,HL22,HV22)
      SS22=ENTROP(T22,VV22,XV22)
      DIFH=(1.+DM1)*(H22-H21)
      ELECP=DIFH/EFFMOT
      COOL=ELECP-DIFH
      H8C=H21-(H22-H21)*(1.-EFFMOT)/EFFMOT
      FSUB= H8/H8C
      WRITE(6,459)IDT7,XDT7,XDT7MI,XDT7MA,H8,H8C,FSUB
      IF (ABS(FSUB-1).LE.0.001) GO TO 455
  459 FORMAT(I5,6F12.3)
      CALL CNVRG(XDT7,XDT7MI,XDT7MA,FSUB,1.,IDT7,DUM1,DUM2,6)
      IF (IDT7-30) 456,456,457
  457 STOP
  455 CONTINUE
      CALL HPIN(H22,P22,XMOL,T22,YQ22,XL22,XV22,VL22,VV22
     1,HL22,HV22)
      WRITE(6,*)
      FL1=1.
      FL5=FL1
      FL25=DM1
      FL6=FL1+FL25
      FL7=FL6
      FL8=FL7
      FL21=FL8
      FL22=FL21
      FL4=FL6
C MERGE EVAP SUBCOOL I   (2)
      FL2=1.+DM1
      T2=T1
      P2=P1
      H2=H1
      YQ2=YQ1
C (26)  SUB LOW COLD OUT
      SS26=SS1
C (25)  SUB LOW COLD IN
      SS25=SS5
C (2)  SUB HIGH COLD IN
      SS2=SS1
C CALCULATE FOR DESIGN HEAT LOAD HEATL(TONS)
C 1 TON = 3.52 KJ/SEC = 211.2 KJ/MIN
C EVMOL  EVAP FLOWRATE MOLES/MIN
      QLOAE= HEATL*211.2
      EVMOL=QLOAE/(H1-H5)
      VRATE=FL21*VVCOM*EVMOL
      EE=EVMOL
      CALL CONAME(IU,ISH,TCOMP)
      CALL UNIT(IU,TCOMP, 1,T1,P1,H1,FL1,YQ1,WM,EE,XMOL,SS1)
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```
      CALL UNIT(-IU,TCOMP,26,T1,P1,H1,FL25,YQ1,WM,EE,XMOL,SS26)
      CALL UNIT(IU,TCOMP, 5,T5,P5,H5,FL5,YQ5,WM,EE,XMOL,SS5)
      CALL UNIT(-IU,TCOMP,25,T5,P5,H5,FL25,YQ5,WM,EE,XMOL,SS25)
      CALL UNIT(IU,TCOMP, 2,T2,P2,H2,FL2,YQ2,WM,EE,XMOL,SS2)
      CALL UNIT(IU,TCOMP, 6,T6,P6,H6,FL6,YQ6,WM,EE,XMOL,SS6)
      CALL UNIT(IU,TCOMP, 7,T7,P7,H7,FL7,YQ7,WM,EE,XMOL,SS7)
      CALL UNIT(IU,TCOMP, 4,T4,P4,H4,FL4,YQ4,WM,EE,XMOL,SS4)
      CALL UNIT(IU,TCOMP, 8,T8,P8,H8,FL8,YQ8,WM,EE,XMOL,SS8)
      CALL UNIT(IU,TCOMP,21,T21,P21,H21,FL21,YQ21,WM,EE,XMOL,SS21)
      CALL UNIT(IU,TCOMP,22,T22,P22,H22,FL22,YQ22,WM,EE,XMOL,SS22)
C     SINK OUTLET TEMPERATURE  (5 F PINCH POINT)
      DTPIN=5./1.8
      IF (YQ22.LE.1) THEN
      T42=T22-DTPIN
      ELSE
      CALL BUBLP(P22,XXX,XV22,T22VS,VL22S,VV22S,.FALSE.,LCRIT)
      CALL HCVCPS(1,T22VS,VV22S,XV22,H22S,CV,CP,VS)
      RAT=(H22-H4)/(H22S-H4)
      T42=(T22VS-DTPIN-T41)*RAT+T41
      ENDIF
      CALL TES(IU,1+DM1,T41,T22,T4,T42,0.,H22,H4,0.,0.,P22,P4,0.,XMOL,
     *0.,NDIV,EVMOL,UASIC,-1,3)
      Q41=FL22*(H22-H4)
      FL41=Q41/(T42-T41)/CPP(IFSI)
      Q31=(H1-H5)
      FL31=Q31/(T31-T32)/CPP(IFSO)
      CALL SINSOU(FEOUT,FCOUT,IU,FL31,T31,T32,FL41,T41,T42,EE)
      QEVA1=(H1-H5)/WM
      COP= (H1-H5)/ELECP
      IF (IFSI.EQ.1) RHO=101.325/0.287/T41
      POFAN=FL41*DPSI/RHO/FANEF
      ELECT=POFAN+ELECP
      COPT=(H1-H5)/ELECT
      IF (IU.EQ.0) WRITE(6,125)T5,DTSUPE,QEVA1,VRATE
      IF (IU.EQ.1) WRITE(6,126)T5*1.8-460.,DTSUPE*1.8,
     $.948*QEVA1/2.20462,VRATE*264.15
      WRITE(6,127)COP,COPT,PR1,YQ21
 125  FORMAT(//,30X,'  TEM EVAP IN (K)        =',F10.1,
     &/,        30X,'  DT SUPERHEAT (K)       =',F10.1,
     &/,        30X,'  COOLING RATE (KJ/KG)   =',F10.1,
     &/,        30X,'  SUC VOL RATE(M**3/MIN) =',F10.1)
 126  FORMAT(//,30X,'  TEM EVAP IN (F)        =',F10.1,
     &/,        30X,'  DT SUPERHEAT (F)       =',F10.1,
     &/,        30X,'  COOLING RATE (BTU/LBM) =',F10.1,
     &/,        30X,'  SUC VOL RATE(M**3/MIN) =',F10.1)
 127  FORMAT(
     &          30X,'  COP                    =',F10.2,
     &/,        30X,'  COPT                   =',F10.2,
     &/,        30X,'  PRESSURE RATIO         =',F10.2,
     &/,        30X,'  QUALITY M/C OUT        =',F10.2,//)
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C IRREVERSIBILITY ANALYSIS
      IRRE=1
      IF (IRRE.EQ.1) THEN
      WRITE(6,*)'     IRREVERSIBILITY ANALYSIS '
      IF (IU.EQ.0) WRITE(6,700)TIRR
```

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      IF (IU.EQ.1) WRITE(6,701)TIRR*1.8-460.
  700 FORMAT(6X,' AT INLET SINK TEMPERATURE (K) =',F8.1,/,
     &5X,'FLOW RATE      ENTROPY           IRRE.',/,
     &5X,' KG/MIN         KJ/KG-K                ',/)
  701 FORMAT(6X,' AT INLET SINK TEMPERATURE (F) =',F8.1,/,
     &5X,'FLOW RATE      ENTROPY           IRRE.',/,
     &5X,' LBM/MIN      BTU/LBM-R              ',/)
      WRITE(6,*)
C VALVE 6-5
      CALL FIRRE2(IU,'VALVE 6-5      ',TIRR,FL5,SS6,SS5,WM,EE,0.,0.
     $,TG1)
C VALVE 6-25
      CALL FIRRE2(IU,'VALVE 6-25     ',TIRR,FL25,SS6,SS25,WM,EE,0.,0.
     $,TG2)
C M/C 8-21
      CALL FIRREC(IU,'M/C 8-21       ',TIRR,.5*(T21+T22),FL8,H8,H21,
     $SS8,SS21,WM,EE,0.,0.,TG3)
C COMP 21-22
      CALL FIRRE2(IU,'COMP 21-22     ',TIRR,FL21,SS21,SS22,WM,EE,0.,
     $0.,TG4)
C COND 22-4 & 41-42
      CALL FIRRE4(IU,'COND 22-4,41-42',TIRR,FL22,H22,H4,SS22,SS4,WM,
     *FL41,T41,T42,1.,EE,IFSI,TG5)
C EVAP 5-1 & 31-32
      CALL FIRRE4(IU,'EVAP 5-1,31-32 ',TIRR,FL5,H5,H1,SS5,SS1,WM,
     *FL31,T31,T32,1.,EE,IFSO,TG6)
C SUB1 4-7 & 2-8
      CALL FIRRE4(IU,'SUB1 4-7,2-8   ',TIRR,FL4,H4,H7,SS4,SS7,WM,
     *FL2,SS2,SS8,WM,EE,0,TG7)
C SUB2 7-6 & 25-26
      CALL FIRRE4(IU,'SUB2 7-6,25-26 ',TIRR,FL7,H7,H6,SS7,SS6,WM,
     *FL25,SS25,SS26,WM,EE,0,TG8)
      TGTOT=TG1+TG2+TG3+TG4+TG5+TG6+TG7+TG8
C
      WRITE(6,447) TGTOT
      ENDIF
 447  FORMAT('*************** TOTAL = ',F10.2,//)
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
      IF (ICHECK.EQ.0) GOTO 130
      IF (IPRI.EQ.1) WRITE(6,*)' HX LOW 1'
      CALL TES(IU,1+DM1,T5,T7,T6,T1,H5,H7,H6,H1,P5,P7,P6,P1,XMOL,XMOL,
     *NDIV,EVMOL,UASI1,IPRI,1)
      IF (IPRI.EQ.1) WRITE(6,*)' HX HIGH 2'
      CALL TES(IU,1+DM1,T2,T4,T7,T8,H2,H4,H7,H8,P2,P4,P7,P8,XMOL,XMOL,
     *NDIV,EVMOL,UASI2,IPRI,2)
      IF (IPRI.EQ.1) WRITE(6,*)' CONDENSER 3'
      CALL TES(IU,1+DM1,T41,T22,T4,T42,0.,H22,H4,0.,0.,P22,P4,0.,XMOL,
     *0.,NDIV,EVMOL,UASIC,IPRI,3)
      IF (IPRI.EQ.1)WRITE(6,*)' EVAPORATOR 4'
      CALL TES(IU,0.,T5,T31,T32,T1,H5,0.,0.,H1,P5,0.,0.,P1,0.,
     *XMOL,NDIV,EVMOL,UASIE,IPRI,4)
      UATOTT=UASI1+UASI2+UASIC+UASIE
      IF (IU.EQ.0) WRITE(90,925)T4,T5,DTSUPE,COP,COPT,PR1,FCOUT,FEOUT,
     *UASIC,UASIE,UASI2,UASI1,UATOTT,VRATE
      IF (IU.EQ.1)  WRITE(90,925)T4*1.8-460.,T5*1.8-460.,DTSUPE*1.8,COP,
     *COPT,PR1,FCOUT,FEOUT,UASIC*.5267,UASIE*.5267,UASI2*.5267,
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     *UASI1*.5267,UATOTT*.5267,VRATE*264.15
 925  FORMAT(1X,2F5.1,F6.1,3F5.2,F6.1,6F5.1,F4.0)
 130  CONTINUE
      RETURN
      END
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C                     CYCLE 6 /NARM
C              TWO PHASE OUTLET AT EVAPORATOR WITH LIQUID PUMP
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
      SUBROUTINE VCS2(ISH,IPRI,IDIAG,IW,FLUD1,FLUD2,VCSN,X,F0,F1,
     *IU,NDIV,ICHECK,IFSI,IFSO,HEATL,T5,DTD,DTSUPE,DPEVA,
     *DTHX,DPHX1,DPHX2,DPHX,T4,DPCO,DPCOOL,DTSUB,PR1I,TIME,DPR,
     *COMPEF,EFFMOT,DTSUP2,PUMPEF,FANEF,DPSI,T31,T32,T41,T42,COP)
C
C
      IMPLICIT REAL*8 (A-H,O-Z)
      CHARACTER*6 HREF(0:20),FLUD1,FLUD2,VCSN
      CHARACTER*20 TITC
      CHARACTER*17 TCOMP(50)
      LOGICAL LBUB,LCRIT
      COMMON /ESDATA/ COEFF(9,20),CRIT(5,20)
      COMMON /HREF1/ HREF,MMAX
      COMMON /HSZERO/ HZERO(20),SZERO(20)
      COMMON /TOLSH/ TOLH,TOLS
      COMMON /RDATA4/ R
      DIMENSION TT(20),PP(20),HH(20),SS(20),TLA(200),THA(200),DELT(200)
      DIMENSION CPP(2)
      DATA CPP/1.0,4.18/
      DATA IFIRST/0/
      DATA TOLH,TOLS/0.00001,0.00001/
C
C                    ****** NOTES *****
C
C  THE QUALITY CALCULATED BY THE NBS ROUTINES IS A MOLAR QUALITY
C
      IR     = 80
      IW     = 6
C
C
C  INITIALIZE COMMON BLOCKS FOR PURE COMPONENTS
C
      DO 100 I = 1,20
         IF (FLUD1 .EQ. HREF(I)) IR1 = I
         IF (FLUD2 .EQ. HREF(I)) IR2 = I
 100  CONTINUE
C
      CALL BCONST(IR1,IR2,F0,F1)
      WM1=CRIT(1,IR1)
      WM2=CRIT(1,IR2)
      XMOL=X
      WM=XMOL*WM1+(1.0-XMOL)*WM2
C     TIRR DEAD ZONE TEMOERATURE AS A BASE TEMPERATURE TO CALCULATE
```

```
C     IRREVERSIBILITY
      TIRR=298.
C
      T6= T5 + DTD
      LBUB = .TRUE.
C
C  AT T6 AND SATURATED LIQUID CONDITION, OBTAIN H6
C
      CALL BUBLT(T6,XMOL,XV6,P6S,VL6S,VV6S,LBUB,LCRIT)
      CALL HCVCPS(1,T6,VL6S,XMOL,H6,CP,CP,VS)
      H5 = H6
C
C  AT H5 AND T5, OBTAIN P5,YQ5,VL5,ETC, FOR TWO PHASE MIXTURE
C
      CALL HTIN(H5,T5,XMOL,P5,YQ5,XL5,XV5,VL5,VV5,HL5,HV5)
C(5)  ENTROPY INLET TO EVAPORATOR
      SL5=ENTROP(T5,VL5,XL5)
      SV5=ENTROP(T5,VV5,XV5)
      S5=YQ5*SV5+(1.-YQ5)*SL5
      SS5=S5
C
C  AT P1  OBTAIN T1,YQ1,VL1,ETC.
C
      P1=P5-DPEVA
      CALL BUBLP(P1,XML,XMOL,T1V,VL1,VV1,.FALSE.,LCRIT)
      T1=T1V+DTSUPE
      HMIN = H5
      HMAX = HV5*2.
      CALL PTIN(1,HMIN,HMAX,P1,T1,XMOL,H1,YQ1,XL1,XV1,
     1              VL1,VV1,HL1,HV1)
C (1) ENTROPY EVAPORATOR OUTLET
      SL1 = ENTROP(T1,VL1,XL1)
      SV1 = ENTROP(T1,VV1,XV1)
      IF (YQ1.LE.0.) S1=SL1
      IF (YQ1.GT.0..AND.YQ1.LT.1.) S1= YQ1*SV1 + (1.0-YQ1)*SL1
      IF (YQ1.GT.1.0)S1=SV1
      SS1=S1
      P8=P1-DPHX
      ITIME=TIME
      DO 130 IPR=1,ITIME
      PR1=PR1I+DPR*(IPR-1)
      P22=PR1*P8
      P4=P22-DPCO
C
C  AT T4 AND P4 CONDITION OBTAIN H4,ETC
C
      HMIN=H5
      HMAX=3*H1
      CALL PTIN(1,HMIN,HMAX,P4,T4,XMOL,H4,YQ4,XL4,XV4,
     1              VL4,VV4,HL4,HV4)
C (4) ENTROPY CONDENSER OUTLET
       SL4 = ENTROP(T4,VL4,XL4)
       SV4 = ENTROP(T4,VV4,XV4)
      IF (YQ4.LE.0.) S4=SL4
      IF (YQ4.GT.0..AND.YQ4.LT.1.) S4= YQ4*SV4 + (1.0-YQ4)*SL4
      IF (YQ4.GT.1.0)S4=SV4
```

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      SS4=S4
C   PRESSURE AT STATE POINT 4,7,6
      P7=P4-DPHX1
      P6=P7-DPHX2
      CALL HPIN(H6,P6,XMOL,T6,YQ6,XL6,XV6,VL6,VV6,HL6,HV6)
C (6) ENTROPY SUBLOW HOT OUTLET
       SL6 = ENTROP(T6,VL6,XL6)
       SV6 = ENTROP(T6,VV6,XV6)
      IF (YQ6.LE.0.) S6=SL6
      IF (YQ6.GT.0..AND.YQ6.LT.1.) S6=YQ6*SV6 +(1.0-YQ6)*SL6
      IF (YQ6.GE.1.0) S6=SV6
      SS6=S6
CT7=T1+DTHX
        CALL BUBLP(P7,XMOL,XV7,T7,VL7,VV7,.TRUE.,LCRIT)
        CALL HCVCPS(1,T7,VL7,XMOL,H7,CP,CP,VS)
C (7) C (7) ENTROPY SUB LOW HOT INLET
        YQ7=0.0
        S7 = ENTROP(T7,VL7,XMOL)
      SS7=S7
C    CALCULATE FLOWRATE FOR SUB II BASE 1 MOLE OF EVAP FLOWRATE)
      DM1=(H5-H7)/(H7-H1)
      H8=H4-H7+H1
      CALL HPIN(H8,P8,XMOL,T8,YQ8,XL8,XV8,VL8,VV8,HL8,HV8)
C (10) ENTROPY PUMP IN
      SL8 = ENTROP(T8,VL8,XL8)
      SS10=SL8
C (21) ENTROPY COMPRESSOR IN
      SV8 = ENTROP(T8,VV8,XV8)
      VVCOM=VV8
      SS21=SV8
C (8)  ENTROPY SEPARATOR IN OR SUB HIGH OUT
      SS8= YQ8*SV8 + (1.0-YQ8)*SL8
C
      WML8= XL8*WM1 + (1.-XL8)*WM2
      WMV8= XV8*WM1 + (1.-XV8)*WM2
      DMV8= (1.+DM1)*YQ8
      DML8= 1.+DM1-DMV8
C 21 VAPOR OUT FROM SEPERATOR
      YQ21=1.
      P21=P8
      T21=T8
      S21=SV8
      H21=HV8
C COMPRESSOR
      CALL SPIN(S21,P22,XV8,T22S,YQ22,XL22,XV22,VL22,VV22,
     1SL22,SV22)
      CALL HCVCPS(1,T22S,VV22,XV22,H22S,CV,CP,VS)
      H22 = (H22S-H21)/COMPEF + H21
      CALL HPIN(H22,P22,XV8,T22,YQ22,XL22,XV22,VL22,VV22
     1,HL22,HV22)
      SS22=ENTROP(T22,VV22,XV22)
      DIFC=DMV8*(H22-H21)
C LIQUID PUMP
      P14=P22
      P9=P14+DPCOOL
      S9=SL8
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```
      CALL SPIN(S9,P9,XL8,T9S,XQ9,XL8,XV9,VL9,VV9,SL9,SV9)
      CALL HCVCPS(1,T9S,VL9,XL8,H9S,CV,CP,VS)
      H9 = (H9S-HL8)/PUMPEF + HL8
      CALL HPIN(H9,P9,XL8,T9,YQ9,XL9,XV9,VL9,VV9,HL9,HV9)
C (9) ENTROPY PUMP OUTLET
      SS9=ENTROP(T9,VL9,XL9)
      DIFP= DML8*(H9-HL8)
      DIFH=DIFC+DIFP
      ELECP=DIFH/EFFMOT
      COOL=ELECP-DIFH
      H14= H9+COOL/DML8
      CALL HPIN(H14,P14,XL8,T14,YQ14,XL14,XV14,VL14,VV14
     1,HL14,HV14)
C (14) ENTROPY M/C OUTLET
       SL14 = ENTROP(T14,VL14,XL14)
       SV14 = ENTROP(T14,VV14,XV14)
      IF (YQ14.LE.0.) S14=SL14
      IF (YQ14.GT.0..AND.YQ14.LT.1.) S14=YQ14*SV14 +(1.0-YQ14)*SL14
      IF (YQ14.GT.1.0)S14=SV14
      SS14=S14
      H23= (H14*DML8+H22*DMV8)/(1+DM1)
      P23=P22
      CALL HPIN(H23,P23,XMOL,T23,YQ23,XL23,XV23,VL23,VV23
     1,HL23,HV23)
C (23) ENTROPY CONDENSER INLET
       SL23 = ENTROP(T23,VL23,XL23)
       SV23 = ENTROP(T23,VV23,XV23)
      IF (YQ23.LE.0.) S23=SL23
      IF (YQ23.GT.0..AND.YQ23.LT.1.) S23=YQ23*SV23 +(1.0-YQ23)*SL23
      IF (YQ23.GT.1.0)S23=SV23
      SS23=S23
      IF (IPRI.EQ.1) THEN
      WRITE(6,*)
      WRITE(6,*) '********* PR COMP = ',PR1
      WRITE(6,*) '********* PR PUMP = '
      ENDIF
      FL1=1.
      FL5=FL1
      FL25=DM1
      FL6=FL1+FL25
      FL7=FL6
      FL8=FL7
      FL23=FL6
      FL21=DMV8
      FL22=FL21
      FL4=FL6
      FL14=DML8
      FL9=FL14
C MERGE EVAP SUBCOOL I   (2)
      FL2=1.+DM1
      T2=T1
      P2=P1
      H2=H1
      YQ2=YQ1
C LIQUID PUMP OUT  (10)
      FL10=FL9
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```
      P10=P8
      T10=T8
      H10=HL8
      YQ10=0.
C (25) SUBCOOLER LOW INLET
      SS25=SS5
C (26) SUBCOOLER LOW OUTLET
      SS26=SS1
C  (2) SUBCOOLER LOW COLD INLET
      SS2=SS1
C CALCULATE FOR DESIGN HEAT LOAD HEATL(TONS)
C 1 TON = 3.52 KJ/SEC = 211.2 KJ/MIN
C EVMOL  EVAP FLOWRATE MOLES/MIN
      QLOAE= HEATL*211.2
      EVMOL=QLOAE/(H1-H5)
      VRATE=FL21*VVCOM*EVMOL
      EE=EVMOL
      IF (IPRI.EQ.0) GOTO 770
      CALL CONAME(IU,ISH,TCOMP)
      CALL UNIT(IU,TCOMP, 1,T1,P1,H1,FL1,YQ1,WM,EE,XMOL,SS1)
      CALL UNIT(-IU,TCOMP,26,T1,P1,H1,FL25,YQ1,WM,EE,XMOL,SS26)
      CALL UNIT(IU,TCOMP, 5,T5,P5,H5,FL5,YQ5,WM,EE,XMOL,SS5)
      CALL UNIT(-IU,TCOMP,25,T5,P5,H5,FL25,YQ5,WM,EE,XMOL,SS25)
      CALL UNIT(IU,TCOMP, 2,T2,P2,H2,FL2,YQ2,WM,EE,XMOL,SS2)
      CALL UNIT(IU,TCOMP, 6,T6,P6,H6,FL6,YQ6,WM,EE,XMOL,SS6)
      CALL UNIT(IU,TCOMP, 7,T7,P7,H7,FL7,YQ7,WM,EE,XMOL,SS7)
      CALL UNIT(IU,TCOMP, 4,T4,P4,H4,FL4,YQ4,WM,EE,XMOL,SS4)
      CALL UNIT(IU,TCOMP, 8,T8,P8,H8,FL8,YQ8,WM,EE,XMOL,SS8)
      CALL UNIT(IU,TCOMP,10,T10,P10,H10,FL10,YQ10,WML8,EE,XL8,SS10)
      CALL UNIT(IU,TCOMP, 9,T9,P9,H9,FL9,YQ9,WML8,EE,XL8,SS9)
      CALL UNIT(IU,TCOMP,14,T14,P14,H14,FL14,YQ14,WML8,EE,XL8,SS14)
      CALL UNIT(IU,TCOMP,21,T21,P21,H21,FL21,YQ21,WMV8,EE,XV8,SS21)
      CALL UNIT(IU,TCOMP,22,T22,P22,H22,FL22,YQ22,WMV8,EE,XV8,SS22)
      CALL UNIT(IU,TCOMP,23,T23,P23,H23,FL23,YQ23,WM,EE,XMOL,SS23)
 770  CONTINUE
C     SINK OUTLET TEMPERATURE  (5 F PINCH POINT)
      DTPIN=5./1.8
      IF (YQ23.LE.1) THEN
      T42=T23-DTPIN
      ELSE
      CALL BUBLP(P23,XXX,XV23,T23VS,VL23S,VV23S,.FALSE.,LCRIT)
      CALL HCVCPS(1,T23VS,VV23S,XV23,H23S,CV,CP,VS)
      RAT=(H23-H4)/(H23S-H4)
      T42=(T23VS-DTPIN-T41)*RAT+T41
      ENDIF
      CALL TES(IU,1+DM1,T41,T23,T4,T42,0.,H23,H4,0.,0.,P23,P4,0.,XMOL,
     *0.,NDIV,EVMOL,UASIC,-1,3)
      Q41=FL23*(H23-H4)
      FL41=Q41/(T42-T41)/CPP(IFSI)
      Q31=(H1-H5)
      FL31=Q31/(T31-T32)/CPP(IFSO)
      CALL SINSOU(FEOUT,FCOUT,IU,FL31,T31,T32,FL41,T41,T42,EE)
      QEVA1=(H1-H5)/WM
      COP= (H1-H5)/ELECP
      IF (IFSI.EQ.1) RHO=101.325/0.287/T41
      POFAN=FL41*DPSI/RHO/FANEF
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      ELECT=POFAN+ELECP
      COPT=(H1-H5)/ELECT
      IF (IU.EQ.0) WRITE(6,125)T5,DTSUPE,QEVA1,VRATE
      IF (IU.EQ.1) WRITE(6,126)T5*1.8-460.,DTSUPE*1.8,
     $.948*QEVA1/2.20462,VRATE*264.15
      WRITE(6,127)COP,COPT,PR1,YQ14
 125  FORMAT(//,30X,'  TEM EVAP IN (K)        =',F10.1,
     &/,        30X,'  DT SUPERHEAT (K)       =',F10.1,
     &/,        30X,'  COOLING RATE (KJ/KG)   =',F10.1,
     &/,        30X,'  SUC VOL RATE(M**3/MIN) =',F10.1)
 126  FORMAT(//,30X,'  TEM EVAP IN (F)        =',F10.1,
     &/,        30X,'  DT SUPERHEAT (F)       =',F10.1,
     &/,        30X,'  COOLING RATE (BTU/LBM) =',F10.1,
     &/,        30X,'  SUC VOL RATE (GPM)     =',F10.1)
 127  FORMAT(
     &          30X,'  COP                    =',F10.2,
     &/,        30X,'  COPT                   =',F10.2,
     &/,        30X,'  PRESSURE RATIO         =',F10.2,
     &/,        30X,'  QUALITY M/C OUT        =',F10.2,//)
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C IRREVERSIBILITY ANALYSIS
      WRITE(6,*)'     IRREVERSIBILITY ANALYSIS '
      WRITE(6,*)'      AT TEMPERATURE',TIRR
      WRITE(6,*)
C VALVE 6-5
      CALL FIRRE2(IU,'VALVE 6-5      ',TIRR,FL5,SS6,SS5,WM,EE,0.,0.
     $,TG1)
C VALVE 6-25
      CALL FIRRE2(IU,'VALVE 6-25     ',TIRR,FL25,SS6,SS25,WM,EE,0.,0.
     $,TG2)
C M/C 9-14
      CALL FIRREC(IU,'M/C 9-14       ',TIRR,.5*(T21+T22),FL9,H9,H14,
     $SS9,SS14,WML8,EE,0.,0.,TG3)
C COMP 21-22
      CALL FIRRE2(IU,'COMP 21-22     ',TIRR,FL21,SS21,SS22,WM,EE,0.,
     $0.,TG4)
C PUMP 10-9
      CALL FIRRE2(IU,'PUMP 10-9      ',TIRR,FL10,SS10,SS9,WML8,EE,0.,
     $0.,TG5)
C COND 23-4 & 41-42
      CALL FIRRE4(IU,'COND 23-4,41-42',TIRR,FL23,H23,H4,SS23,SS4,WM,
     *FL41,T41,T42,1.,EE,IFSI,TG6)
C EVAP 5-1 & 31-32
      CALL FIRRE4(IU,'EVAP 5-1,31-32 ',TIRR,FL5,H5,H1,SS5,SS1,WM,
     *FL31,T31,T32,1.,EE,IFSO,TG7)
C SUB1 4-7 & 2-8
      CALL FIRRE4(IU,'SUB1 4-7,2-8   ',TIRR,FL4,H4,H7,SS4,SS7,WM,
     *FL2,SS2,SS8,WM,EE,0,TG8)
C SUB2 7-6 & 25-26
      CALL FIRRE4(IU,'SUB2 7-6,25-26 ',TIRR,FL7,H7,H6,SS7,SS6,WM,
     *FL25,SS25,SS26,WM,EE,0,TG9)
C SEPA 8-10-21
      CALL FIRRE3(IU,'SEPA 8-10-21   ',TIRR,FL8,SS8,WM,FL10,SS10,WML8,
     *FL21,SS21,WMV8,EE,-1,TG10)
C MIX 14-22-23
      CALL FIRRE3(IU,'MIX 14-22-23   ',TIRR,FL14,SS14,WML8,FL22,SS22,
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     *WMV8,FL23,SS23,WM,EE,1,TG11)
      TGTOT=TG1+TG2+TG3+TG4+TG5+TG6+TG7+TG8+TG9+TG10+TG11
C
      WRITE(6,447) TGTOT
 447  FORMAT('*************** TOTAL = ',F10.2,//)
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
      IF (ICHECK.EQ.0) GOTO 130
C     IF (IU.EQ.1) WRITE(6,6010)
C     IF (IU.EQ.0) WRITE(6,6011)
C6010 FORMAT(5X,'HEAT LOAD    PINCH POINT     LMTD       UA',/,
C    *5X,        ' BTU/MIN        F            F       BTU/MIN-F',/)
C6011   FORMAT(5X,'HEAT LOAD    PINCH POINT     LMTD       UA',/,
C    *5X,        ' KJ/MIN         K            K      KJ/MIN-K',/)
      IF (IPRI.EQ.1)  WRITE(6,*)' HX LOW  1'
      CALL TES(IU,1+DM1,T5,T7,T6,T1,H5,H7,H6,H1,P5,P7,P6,P1,XMOL,XMOL,
     *NDIV,EVMOL,UASI1,IPRI,1)
      IF (IPRI.EQ.1)  WRITE(6,*)' HX HIGH 2'
      CALL TES(IU,1+DM1,T2,T4,T7,T8,H2,H4,H7,H8,P2,P4,P7,P8,XMOL,XMOL,
     *NDIV,EVMOL,UASI2,IPRI,2)
       WRITE(6,*)
      WRITE(6,*)' CONDENSER  3'
      WRITE(6,*)
      CALL TES(IU,1+DM1,T41,T23,T4,T42,0.,H23,H4,0.,0.,P23,P4,0.,XMOL,
     *0.,NDIV,EVMOL,UASIC,IPRI,3)
      IF (IPRI.EQ.1) WRITE(6,*)' EVAPORATOR  4'
      CALL TES(IU,0.,T5,T31,T32,T1,H5,0.,0.,H1,P5,0.,0.,P1,0.,
     *XMOL,NDIV,EVMOL,UASIE,IPRI,4)
      UATOTT=UASI1+UASI2+UASIC+UASIE
      IF (IU.EQ.0) WRITE(90,925)T4,T5,DTSUPE,COP,COPT,PR1,
     *FCOUT,FEOUT,UASIC,UASIE,UASI1,UASI2,UATOTT,YQ1,YQ14,VRATE
      IF (IU.EQ.1)  WRITE(90,925)T4*1.8-460.,T5*1.8-460.,DTSUPE*1.8,
     *COP,COPT,PR1,FCOUT,FEOUT,UASIC*.5267,UASIE*.5267,
     *UASI1*.5267,UASI2*.5267,UATOTT*.5267,YQ1,YQ14,VRATE*264.15
 925  FORMAT(1X,3F5.1,3F5.2,F6.1,6F5.1,2F4.2,F4.0)
 130  CONTINUE
      RETURN
      END
C
CCCCCCCCCCCCCCCCCCCC
```

```
      COMMON /ESDATA/ COEFF(9,20),CRIT(5,20)
      COMMON /HREF1/ HREF,MMAX
      COMMON /HSZERO/ HZERO(20),SZERO(20)
      COMMON /TOLSH/ TOLH,TOLS
      COMMON /RDATA4/ R
      DIMENSION TT(20),PP(20),HH(20),SS(20),TLA(200),THA(200),DELT(200)
      DIMENSION CPP(2)
      DATA CPP/1.0,4.18/
      DATA IFIRST/0/
      DATA TOLH,TOLS/0.00001,0.00001/
C
C                    ****** NOTES *****
C
C  THE QUALITY CALCULATED BY THE NBS ROUTINES IS A MOLAR QUALITY
C
      IR     = 80
      IW     = 6
C
C
C  INITIALIZE COMMON BLOCKS FOR PURE COMPONENTS
C
      DO 100 I = 1,20
         IF (FLUD1 .EQ. HREF(I)) IR1 = I
         IF (FLUD2 .EQ. HREF(I)) IR2 = I
 100  CONTINUE
C
      CALL BCONST(IR1,IR2,F0,F1)
      WM1=CRIT(1,IR1)
      WM2=CRIT(1,IR2)
      XMOL=X
      WM=XMOL*WM1+(1.0-XMOL)*WM2
C     TIRR DEAD ZONE TEMOERATURE AS A BASE TEMPERATURE TO CALCULATE
C     IRREVERSIBILITY
      TIRR=298.
C
      T6= T5 + DTD
      LBUB = .TRUE.
C
C  AT T6 AND SATURATED LIQUID CONDITION, OBTAIN H6
C
      CALL BUBLT(T6,XMOL,XV6,P6S,VL6S,VV6S,LBUB,LCRIT)
      CALL HCVCPS(1,T6,VL6S,XMOL,H6,CP,CP,VS)
      H5 = H6
C
C  AT H5 AND T5, OBTAIN P5,YQ5,VL5,ETC, FOR TWO PHASE MIXTURE
C
      CALL HTIN(H5,T5,XMOL,P5,YQ5,XL5,XV5,VL5,VV5,HL5,HV5)
C    ENTROPY INLET TO EVAPORATOR
      SL5=ENTROP(T5,VL5,XL5)
      SV5=ENTROP(T5,VV5,XV5)
      S5=YQ5*SV5+(1.-YQ5)*SL5
      SS5=S5
C
C  AT T4 AND SATURATED LIQUID CONDITION OBTAIN H4,P4,ETC
C
      LBUB = .TRUE.
```

```
      CALL BUBLT(T4,XMOL,XV4,P4,VL4,VV4,LBUB,LCRIT)
      SL4=ENTROP(T4,VL4,XMOL)
C (4) CONDENSER OUTLET
      SS4=SL4
      CALL HCVCPS(1,T4,VL4,XMOL,H4,CP,CP,VS)
      YQ4= 0.
C   PRESSURE AT STATE POINT 4,7,6
      P7=P4-DPHX1
      P6=P7-DPHX2
      CALL HPIN(H6,P6,XMOL,T6,YQ6,XL6,XV6,VL6,VV6,HL6,HV6)
C (6) ENTROPY SUBLOW HOT OUTLET
       SL6 = ENTROP(T6,VL6,XL6)
      SS6=SL6
C
C  AT P1  OBTAIN T1,YQ1,VL1,ETC.
C
      P1=P5-DPEVA
      CALL BUBLP(P1,XML,XMOL,T1V,VL1,VV1,.FALSE.,LCRIT)
      T1=T1V+DTSUPE
      HMIN = H5
      HMAX = HV5*2.
      CALL PTIN(1,HMIN,HMAX,P1,T1,XMOL,H1,YQ1,XL1,XV1,
     1              VL1,VV1,HL1,HV1)
       SL1 = ENTROP(T1,VL1,XL1)
       SV1 = ENTROP(T1,VV1,XV1)
      VVCOM=VV1
      IF (YQ1.LE.0.) S1=SL1
      IF (YQ1.GT.0..AND.YQ1.LT.1.) S1= YQ1*SV1 + (1.0-YQ1)*SL1
      IF (YQ1.GT.1.0)S1=SV1
      SS1=S1
      T7=T1+DTHX
      CALL BUBLT(T7,XMOL,XV7,P7S,VL7S,VV7S,.TRUE.,LCRIT)
      CALL HCVCPS(1,T7,VL7S,XMOL,H7,CP,CP,VS)
      CALL HPIN(H7,P7,XMOL,T7,YQ7,XL7,XV7,VL7,VV7,HL7,HV7)
C (7) ENTROPY SUB LOW HOT INLET
       SL7 = ENTROP(T7,VL7,XL7)
       SS7=SL7
C (27) ENTROPY SUB HIGH HOT OUTLET
      SS27=SS7
C    CALCULATE FLOWRATE FOR SUB II BASE 1 MOLE OF EVAP FLOWRATE)
      DM1=(H5-H7)/(H7-H1)
C
C    FIRST STAGE OF COMPRESSOR
      ITIME=TIME
      DO 130 IPR=1,ITIME
      PR1=PR1I+DPR*(IPR-1)
      P2=P1*PR1
      CALL SPIN(S1,P2,XMOL,T2S,YQ2,XL2,XV2,VL2,VV2,SL2,SV2)
      CALL HCVCPS(1,T2S,VV2,XV2,H2S,CV,CP,VS)
      H2 = (H2S-H1)/COMPEF + H1
      CALL HPIN(H2,P2,XMOL,T2,YQ2,XL2,XV2,VL2,VV2,HL2,HV2)
      SS2=ENTROP(T2,VV2,XV2)
C ITERATION FOR FLOW RATE HX HIGH
      P22=P4+DPCO
      P21=P2
      P20=P21
```

```
      P19=P20+DPCOOL
      P18=P19+DPHX
      H18=H7
      PR2=P22/P21
      DM2=0.2
      DM2MA=10.
      DM2MI=0.001
      IDM=0
  140 H19=(1.+DM1+DM2)*(H4-H7)/DM2+H18
      CALL BUBLP(P21,XML,XMOL,T21V,VL21,VV21,.FALSE.,LCRIT)
      T21=T21V+DTSUP2
      HMIN =0.5*H2
      HMAX =H2*1.5
      CALL PTIN(1,HMIN,HMAX,P21,T21,XMOL,H21,YQ21,XL21,XV21,
     1             VL21,VV21,HL21,HV21)
       SL21 = ENTROP(T21,VL21,XL21)
       SV21 = ENTROP(T21,VV21,XV21)
      VVCOM2=VV21
      IF (YQ21.LE.0.) S21=SL21
      IF (YQ21.GT.0..AND.YQ21.LT.1.)S21= YQ21*SV21 + (1.0-YQ21)*SL21
      IF (YQ21.GT.1.0)S21=SV21
      SS21=S21
      CALL SPIN(S21,P22,XMOL,T22S,YQ22,XL22,XV22,VL22,VV22,
     1SL22,SV22)
      CALL HCVCPS(1,T22S,VV22,XV22,H22S,CV,CP,VS)
      H22 = (H22S-H21)/COMPEF + H21
      CALL HPIN(H22,P22,XMOL,T22,YQ22,XL22,XV22,VL22,VV22
     1,HL22,HV22)
      SS22=ENTROP(T22,VV22,XV22)
      DIFH=(1.+DM1)*(H2-H1)+(1.+DM1+DM2)*(H22-H21)
      ELECP=DIFH/EFFMOT
      COOL=ELECP-DIFH
      H20=((1.+DM1+DM2)*H21-(1.+DM1)*H2)/DM2
      H19P=H20-COOL/DM2
      IF (ABS(H19P/H19-1.).LE.0.001) GOTO 160
      CALL CNVRG(DM2,DM2MI,DM2MA,H19,H19P,IDM,DUM3,DUM4,6)
      IF (IDM-30) 140,140,150
  150 STOP
  160 CALL HPIN(H18,P18,XMOL,T18,YQ18,XL18,XV18,VL18,VV18
     1,HL18,HV18)
       SL18 = ENTROP(T18,VL18,XL18)
       SV18 = ENTROP(T18,VV18,XV18)
      IF (YQ18.LE.0.) S18=SL18
      IF (YQ18.GT.0..AND.YQ18.LT.1.)S18= YQ18*SV18 + (1.0-YQ18)*SL18
      IF (YQ18.GT.1.0)S18=SV18
      SS18=S18
      CALL HPIN(H19,P19,XMOL,T19,YQ19,XL19,XV19,VL19,VV19
     1,HL19,HV19)
       SL19 = ENTROP(T19,VL19,XL19)
       SV19 = ENTROP(T19,VV19,XV19)
      IF (YQ19.LE.0.) S19=SL19
      IF (YQ19.GT.0..AND.YQ19.LT.1.)S19= YQ19*SV19 + (1.0-YQ19)*SL19
      IF (YQ19.GT.1.0)S19=SV19
      SS19=S19
      CALL HPIN(H20,P20,XMOL,T20,YQ20,XL20,XV20,VL20,VV20
     1,HL20,HV20)
```

```
       SL20 = ENTROP(T20,VL20,XL20)
       SV20 = ENTROP(T20,VV20,XV20)
      IF (YQ20.LE.0.) S20=SL20
      IF (YQ20.GT.0..AND.YQ20.LT.1.)S20= YQ20*SV20 + (1.0-YQ20)*SL20
      IF (YQ20.GT.1.0)S20=SV20
      SS20=S20
      WRITE(6,*)
      WRITE(6,*)
      WRITE(6,*) '********* PR1 = ',PR1
      WRITE(6,*) '********* PR2 = ',PR2
      FL1=1.
      FL5=FL1
      FL6=FL1+DM1
      FL7=FL6
      FL2=FL6
      FL21=FL6+DM2
      FL22=FL21
      FL4=FL22
      FL18=DM2
      FL19=FL18
      FL20=FL19
      T27=T7
      H27=H7
      YQ27=YQ7
      P27=P7
      FL27=FL4
      FL25=DM1
      FL26=FL25
C (26) SUB LOW COLD OUT
      SS26=SS1
C (25) SUB LOW COLD OUT
      SS25=SS5
C (11) FIRST STAGE COMPRESSOR INLET
      T11=T1
      P11=P1
      H11=H1
      FL11=FL1+FL26
      YQ11=YQ1
      SS11=SS1
C CALCULATE FOR DESIGN HEAT LOAD HEATL(TONS)
C 1 TON = 3.52 KJ/SEC = 211.2 KJ/MIN
C EVMOL  EVAP FLOWRATE MOLES/MIN
      QLOAE= HEATL*211.2
      EVMOL=QLOAE/(H1-H5)
      VRATE=FL11*VVCOM*EVMOL
      VRATE2=FL21*VVCOM2*EVMOL
      EE=EVMOL
      CALL CONAME(IU,ISH,TCOMP)
      CALL UNIT(IU,TCOMP, 1,T1,P1,H1,FL1,YQ1,WM,EE,XMOL,SS1)
      CALL UNIT(IU,TCOMP,26,T1,P1,H1,FL26,YQ1,WM,EE,XMOL,SS26)
      CALL UNIT(IU,TCOMP, 5,T5,P5,H5,FL5,YQ5,WM,EE,XMOL,SS5)
      CALL UNIT(-IU,TCOMP,25,T5,P5,H5,FL25,YQ5,WM,EE,XMOL,SS25)
      CALL UNIT(IU,TCOMP, 6,T6,P6,H6,FL6,YQ6,WM,EE,XMOL,SS6)
      CALL UNIT(IU,TCOMP, 7,T7,P7,H7,FL7,YQ7,WM,EE,XMOL,SS7)
      CALL UNIT(IU,TCOMP, 4,T4,P4,H4,FL4,YQ4,WM,EE,XMOL,SS4)
      CALL UNIT(IU,TCOMP,11,T11,P11,H11,FL11,YQ11,WM,EE,XMOL,SS11)
```

```
      CALL UNIT(IU,TCOMP, 2,T2,P2,H2,FL2,YQ2,WM,EE,XMOL,SS2)
      CALL UNIT(IU,TCOMP, 20,T20,P20,H20,FL20,YQ20,WM,EE,XMOL,SS20)
      CALL UNIT(IU,TCOMP,21,T21,P21,H21,FL21,YQ21,WM,EE,XMOL,SS21)
      CALL UNIT(IU,TCOMP,22,T22,P22,H22,FL22,YQ22,WM,EE,XMOL,SS22)
      CALL UNIT(IU,TCOMP,18,T18,P18,H18,FL18,YQ18,WM,EE,XMOL,SS18)
      CALL UNIT(IU,TCOMP,19,T19,P19,H19,FL19,YQ19,WM,EE,XMOL,SS19)
      CALL UNIT(IU,TCOMP,27,T27,P27,H27,FL27,YQ27,WM,EE,XMOL,SS27)
C     SINK OUTLET TEMPERATURE  (5 F PINCH POINT)
      DTPIN=5./1.8
      IF (YQ22.LE.1) THEN
      T42=T22-DTPIN
      ELSE
      CALL BUBLP(P22,XXX,XV22,T22VS,VL22S,VV22S,.FALSE.,LCRIT)
      CALL HCVCPS(1,T22VS,VV22S,XV22,H22S,CV,CP,VS)
      RAT=(H22-H4)/(H22S-H4)
      T42=(T22VS-DTPIN-T41)*RAT+T41
      ENDIF
      CALL TES(IU,1+DM1+DM2,T41,T22,T4,T42,0.,H22,H4,0.,0.,P22,P4,0.,
     *XMOL,0.,NDIV,EVMOL,UASIC,-1,3)
      Q41=FL22*(H22-H4)
      FL41=Q41/(T42-T41)/CPP(IFSI)
      Q31=(H1-H5)
      FL31=Q31/(T31-T32)/CPP(IFSO)
      CALL SINSOU(FEOUT,FCOUT,IU,FL31,T31,T32,FL41,T41,T42,EE)
      QEVA1=(H1-H5)/WM
      COP= (H1-H5)/ELECP
      IF (IFSI.EQ.1) RHO=101.325/0.287/T41
      POFAN=FL41*DPSI/RHO/FANEF
      ELECT=POFAN+ELECP
      COPT=(H1-H5)/ELECT
      IF (IU.EQ.0) WRITE(6,125)T5,DTHX,QEVA1,VRATE,VRATE2
      IF (IU.EQ.1) WRITE(6,126)T5*1.8-460.,DTHX*1.8,
     $.948*QEVA1/2.20462,VRATE*264.15,VRATE2*264.15
      WRITE(6,127)COP,COPT,PR1,PR2,YQ20
 125  FORMAT(//,30X,'  TEM EVAP IN (K)        =',F10.1,
     &/,        30X,'  DTHX (K)               =',F10.1,
     &/,        30X,'  COOLING RATE (KJ/KG)   =',F10.1,
     &/,        30X,'  SUC1 VOL RATE(M**3/MIN)=',F10.1,
     &/,        30X,'  SUC2 VOL RATE(M**3/MIN)=',F10.1)
 126  FORMAT(//,30X,'  TEM EVAP IN (F)        =',F10.1,
     &/,        30X,'  DTHX (F)               =',F10.1,
     &/,        30X,'  COOLING RATE (BTU/LBM) =',F10.1,
     &/,        30X,'  SUC1 VOL RATE (GPM)    =',F10.1,
     &/,        30X,'  SUC2 VOL RATE (GPM)    =',F10.1)
 127  FORMAT(
     &          30X,'  COP                    =',F10.2,
     &/,        30X,'  COPT                   =',F10.2,
     &/,        30X,'  PRESSURE RATIO 1       =',F10.2,
     &/,        30X,'  PRESSURE RATIO 2       =',F10.2,
     &/,        30X,'  QUALITY M/C OUT        =',F10.2,//)
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C IRREVERSIBILITY ANALYSIS
      WRITE(6,*)'     IRREVERSIBILITY ANALYSIS '
      WRITE(6,*)'      AT TEMPERATURE',TIRR
      WRITE(6,*)
C VALVE 6-5
```

```
      CALL FIRRE2(IU,'VALVE 6-5       ',TIRR,FL5,SS6,SS5,WM,EE,0.,0.
     $,TG1)
C VALVE 6-25
      CALL FIRRE2(IU,'VALVE 6-25      ',TIRR,FL25,SS6,SS25,WM,EE,0.,0.
     $,TG2)
C VALVE 17-18
      CALL FIRRE2(IU,'VALVE 17-18     ',TIRR,FL18,SS27,SS18,WM,EE,0.,0.
     $,TG3)
C M/C 19-20
      CALL FIRREC(IU,'M/C 19-20       ',TIRR,.5*(T11+T22),FL19,H19,H20,
     $SS19,SS20,WM,EE,0.,0.,TG4)
C COMP 11-2
      CALL FIRRE2(IU,'COMP 11-2       ',TIRR,FL2,SS11,SS2,WM,EE,0.,
     $0.,TG5)
C COMP 21-22
      CALL FIRRE2(IU,'COMP 21-22      ',TIRR,FL21,SS21,SS22,WM,EE,0.,
     $0.,TG6)
C COND 22-4 & 41-42
      CALL FIRRE4(IU,'COND 22-4,41-42',TIRR,FL22,H22,H4,SS22,SS4,WM,
     *FL41,T41,T42,1.,EE,IFSI,TG7)
C EVAP 5-1 & 31-32
      CALL FIRRE4(IU,'EVAP 5-1,31-32 ',TIRR,FL5,H5,H1,SS5,SS1,WM,
     *FL31,T31,T32,1.,EE,IFS0,TG8)
C SUB1 4-27 & 18-19
      CALL FIRRE4(IU,'SUB1 4-27,18-19',TIRR,FL4,H4,H27,SS4,SS27,WM,
     *FL18,SS18,SS19,WM,EE,0,TG9)
C SUB2 7-6 & 25-26
      CALL FIRRE4(IU,'SUB2 7-6,25-26 ',TIRR,FL7,H7,H6,SS7,SS6,WM,
     *FL25,SS25,SS26,WM,EE,0,TG10)
C MIX 2-20-21
      CALL FIRRE3(IU,'MIX 2-20-21     ',TIRR,FL2,SS2,WM,FL20,SS20,
     *WM,FL21,SS21,WM,EE,1,TG11)
      TGTOT=TG1+TG2+TG3+TG4+TG5+TG6+TG7+TG8+TG9+TG10+TG11
C
      WRITE(6,447) TGTOT
 447  FORMAT('*************** TOTAL = ',F10.2,//)
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
      IF (ICHECK.EQ.0) GOTO 130
      IF (IPRI.EQ.1) WRITE(6,*)' HX LOW 1'
      CALL TES(IU,1+DM1,T5,T7,T6,T1,H5,H7,H6,H1,P5,P7,P6,P1,XMOL,XMOL,
     *NDIV,EVMOL,UASI1,IPRI,1)
      IF (IPRI.EQ.1) WRITE(6,*)' HX HIGH 2'
      CALL TES(IU,1+DM1+DM2,T18,T4,T7,T19,H18,H4,H7,H19,P18,P4,P7,P19,
     *XMOL,XMOL,NDIV,EVMOL,UASI2,IPRI,2)
      IF (IPRI.EQ.1) WRITE(6,*)' CONDENSER 3'
      CALL TES(IU,1+DM1+DM2,T41,T22,T4,T42,0.,H22,H4,0.,0.,P22,P4,0.,
     *XMOL,0.,NDIV,EVMOL,UASIC,IPRI,3)
      IF (IPRI.EQ.1) WRITE(6,*)' EVAPORATOR 4'
      CALL TES(IU,0.,T5,T31,T32,T1,H5,0.,0.,H1,P5,0.,0.,P1,0.,
     *XMOL,NDIV,EVMOL,UASIE,IPRI,4)
      UATOTT=UASI1+UASI2+UASIC+UASIE
      IF (IU.EQ.0) WRITE(90,925)T4,T5,DTHX,COP,COPT,PR1,PR2,
     *FCOUT,FEOUT,UASIC,UASIE,UASI1,UASI2,UATOTT,VRATE,VRATE2
      IF (IU.EQ.1)  WRITE(90,925)T4*1.8-460.,T5*1.8-460.,DTHX*1.8,
     *COP,COPT,PR1,PR2,FCOUT,FEOUT,UASIC*.5267,UASIE*.5267,
     *UASI1*.5267,UASI2*.5267,UATOTT*.5267,VRATE*264.15,VRATE2*264.15
```

```
 925  FORMAT(1X,3F5.1,4F5.2,F6.1,6F5.1,2F4.0)
 130  CONTINUE
      RETURN
      END
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C                    CYCLE 2 /PURE
C                  TWO-STAGE COMPRESSOR
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
      SUBROUTINE VCSO(ISH,IPRI,IDIAG,IW,FLUD1,FLUD2,VCSN,X,FO,F1,
     *IU,NDIV,ICHECK,IFSI,IFSO,HEATL,T5,DTD,DTSUPE,DPEVA,
     *DTHX,DPHX1,DPHX2,DPHX,T4,DPCO,DPCOOL,DTSUB,PR1I,TIME,DPR,
     *COMPEF,EFFMOT,DTSUP2,PUMPEF,FANEF,DPSI,T31,T32,T41,T42,COP)
      IMPLICIT REAL*8 (A-H,O-Z)
      CHARACTER*6 HREF(0:20),FLUD1,FLUD2,VCSN
      CHARACTER*17 TCOMP(50)
      LOGICAL LBUB,LCRIT
      COMMON /ESDATA/ COEFF(9,20),CRIT(5,20)
      COMMON /HREF1/ HREF,MMAX
      COMMON /HSZERO/ HZERO(20),SZERO(20)
      COMMON /TOLSH/ TOLH,TOLS
      COMMON /RDATA4/ R
      DIMENSION TT(20),PP(20),HH(20),SS(20),TLA(200),THA(200),DELT(200)
      DIMENSION CPP(2)
      DATA CPP/1.0,4.18/
      DATA IFIRST/0/
      DATA TOLH,TOLS/0.00001,0.00001/
C
C                    ****** NOTES *****
C
C  THE QUALITY CALCULATED BY THE NBS ROUTINES IS A MOLAR QUALITY
C
C
C
C  INITIALIZE COMMON BLOCKS FOR PURE COMPONENTS
C
      DO 100 I = 1,20
         IF (FLUD1 .EQ. HREF(I)) IR1 = I
         IF (FLUD2 .EQ. HREF(I)) IR2 = I
 100  CONTINUE
C
      CALL BCONST(IR1,IR2,FO,F1)
      WM1=CRIT(1,IR1)
      WM2=CRIT(1,IR2)
      XMOL=X
      WM=XMOL*WM1+(1.0-XMOL)*WM2
C     TIRR DEAD ZONE TEMOERATURE AS A BASE TEMPERATURE TO CALCULATE
C     IRREVERSIBILITY
      TIRR=298.
C
      LBUB = .TRUE.
C
C  AT T5 AND SATURATED LIQUID CONDITION
```

```
C
      CALL BUBLT(T5,XMOL,XV5,P5,VL5,VV5,LBUB,LCRIT)
C
C  AT T3=T4+DTSUB AND SATURATED LIQUID CONDITION OBTAIN H3,P4,ETC
C
      T3=T4+DTSUB
      LBUB = .TRUE.
      CALL BUBLT(T3,XMOL,XV3,P3,VL3,VV3,LBUB,LCRIT)
      P4=P3
      LBUB = .TRUE.
      CALL BUBLT(T4,XMOL,XV4,P4S,VL4,VV4,LBUB,LCRIT)
      CALL HCVCPS(1,T4,VL4,XMOL,H4,CP,CP,VS)
      CALL HPIN(H4,P4,XMOL,T4S,YQ4,XL4,XV4,VL4,VV4,HL4,HV4)
       SL4 = ENTROP(T4,VL4,XL4)
C (4)  ENTROPY CONDENSER OUT
       SS4=SL4
C
C  AT P1  OBTAIN T1,YQ1,VL1,ETC.
C
      P1=P5-DPEVA
      CALL BUBLP(P1,XML,XMOL,T1V,VL1,VV1,.FALSE.,LCRIT)
      T1=T1V+DTSUPE
      CALL HCVCPS(1,T1V,VV1,XMOL,H1S,CP,CP,VS)
      HMIN = H1S
      HMAX = 1.5*H1S
      CALL PTIN(1,HMIN,HMAX,P1,T1,XMOL,H1,YQ1,XL1,XV1,
     1              VL1,VV1,HL1,HV1)
       SL1 = ENTROP(T1,VL1,XL1)
       SV1 = ENTROP(T1,VV1,XV1)
      VVCOM=VV1
      IF (YQ1.LE.0.) S1=SL1
      IF (YQ1.GT.0..AND.YQ1.LT.1.) S1= YQ1*SV1 + (1.0-YQ1)*SL1
      IF (YQ1.GT.1.0)S1=SV1
      SS1=S1
C   PRESSURE AT STATE POINT 7
      P7=P4-DPHX1
      T7=T1+DTHX
      CALL BUBLT(T7,XMOL,XV7,P7S,VL7,VV7,.TRUE.,LCRIT)
      CALL HCVCPS(1,T7,VL7,XMOL,H7,CP,CP,VS)
      CALL HPIN(H7,P7,XMOL,T7S,YQ7,XL7,XV7,VL7,VV7,HL7,HV7)
C (7) SUBCOOLER OUT
      SL7=ENTROP(T7,VL7,XL7)
      SS7=SL7
      H5=H7
      CALL HPIN(H5,P5,XMOL,T5S,YQ5,XL5,XV5,VL5,VV5,HL5,HV5)
C    ENTROPY INLET TO EVAPORATOR
      SL5=ENTROP(T5,VL5,XL5)
      SV5=ENTROP(T5,VV5,XV5)
      S5=YQ5*SV5+(1.-YQ5)*SL5
      SS5=S5
      DM1=0.
C
C    FIRST STAGE OF COMPRESSOR
      ITIME=TIME
      DO 130 IPR=1,ITIME
      PR1=PR1I+DPR*(IPR-1)
```

```
      P2=P1*PR1
C COMPRESSOR
      CALL SPIN(S1,P2,XMOL,T2S,YQ2,XL2,XV2,VL2,VV2,SL2,SV2)
      CALL HCVCPS(1,T2S,VV2,XV2,H2S,CV,CP,VS)
      H2 = (H2S-H1)/COMPEF + H1
      CALL HPIN(H2,P2,XMOL,T2,YQ2,XL2,XV2,VL2,VV2,HL2,HV2)
      SS2=ENTROP(T2,VV2,XV2)
C ITERATION FOR FLOW RATE HX HIGH
      P22=P4+DPCO
      P21=P2
      P20=P21
      P19=P20+DPCOOL
      P18=P19+DPHX1
      PR2=P22/P21
      H18=H4
      DM2=0.2
      DM2MA=10.
      DM2MI=0.001
      IDM=0
 140  CONTINUE
      H19=(1+DM1)*(H4-H7)/DM2+H18
      CALL BUBLP(P21,XML,XMOL,T21V,VL21,VV21,.FALSE.,LCRIT)
      CALL HCVCPS(1,T21V,VV21,XMOL,H21S,CP,CP,VS)
      T21=T21V+DTSUP2
      HMIN =H21S
      HMAX =H21S*1.5
      CALL PTIN(1,HMIN,HMAX,P21,T21,XMOL,H21,YQ21,XL21,XV21,
     1              VL21,VV21,HL21,HV21)
       SL21 = ENTROP(T21,VL21,XL21)
       SV21 = ENTROP(T21,VV21,XV21)
      VVCOM2=VV21
      IF (YQ21.LE.0.) S21=SL21
      IF (YQ21.GT.0..AND.YQ21.LT.1.)S21= YQ21*SV21 + (1.0-YQ21)*SL21
      IF (YQ21.GT.1.0)S21=SV21
      SS21=S21
      CALL SPIN(S21,P22,XMOL,T22S,YQ22,XL22,XV22,VL22,VV22,
     1SL22,SV22)
      CALL HCVCPS(1,T22S,VV22,XV22,H22S,CV,CP,VS)
      H22 = (H22S-H21)/COMPEF + H21
      CALL HPIN(H22,P22,XMOL,T22,YQ22,XL22,XV22,VL22,VV22
     1,HL22,HV22)
      SS22=ENTROP(T22,VV22,XV22)
      DIFH=(1.+DM1)*(H2-H1)+(1.+DM1+DM2)*(H22-H21)
      ELECP=DIFH/EFFMOT
      COOL=ELECP-DIFH
      H20=((1.+DM1+DM2)*H21-(1.+DM1)*H2)/DM2
      H19P=H20-COOL/DM2
      IF (ABS(H19P/H19-1.).LE.0.001) GOTO 160
      CALL CNVRG(DM2,DM2MI,DM2MA,H19,H19P,IDM,DUM3,DUM4,6)
      IF (IDM-30) 140,140,150
 150  STOP
 160  CALL HPIN(H18,P18,XMOL,T18,YQ18,XL18,XV18,VL18,VV18
     1,HL18,HV18)
       SL18 = ENTROP(T18,VL18,XL18)
       SV18 = ENTROP(T18,VV18,XV18)
      IF (YQ18.LE.0.) S18=SL18
```

```
      IF (YQ18.GT.0..AND.YQ18.LT.1.)S18= YQ18*SV18 + (1.0-YQ18)*SL18
      IF (YQ18.GT.1.0)S18=SV18
      SS18=S18
      CALL HPIN(H19,P19,XMOL,T19,YQ19,XL19,XV19,VL19,VV19
     1,HL19,HV19)
       SL19 = ENTROP(T19,VL19,XL19)
       SV19 = ENTROP(T19,VV19,XV19)
      IF (YQ19.LE.0.) S19=SL19
      IF (YQ19.GT.0..AND.YQ19.LT.1.)S19= YQ19*SV19 + (1.0-YQ19)*SL19
      IF (YQ19.GT.1.0)S19=SV19
      SS19=S19
      CALL HPIN(H20,P20,XMOL,T20,YQ20,XL20,XV20,VL20,VV20
     1,HL20,HV20)
       SL20 = ENTROP(T20,VL20,XL20)
       SV20 = ENTROP(T20,VV20,XV20)
      IF (YQ20.LE.0.) S20=SL20
      IF (YQ20.GT.0..AND.YQ20.LT.1.)S20= YQ20*SV20 + (1.0-YQ20)*SL20
      IF (YQ20.GT.1.0)S20=SV20
      SS20=S20
      WRITE(6,*)
      WRITE(6,*)
      WRITE(6,*) '********* PR1 = ',PR1
      WRITE(6,*) '********* PR2 = ',PR2
      FL1=1.
      FL5=FL1
      FL7=FL1
      FL2=FL1
      FL21=FL1+DM2
      FL22=FL21
      FL3=FL22
      FL4=1.
      FL18=DM2
      FL24=DM2
      FL19=FL18
      FL20=FL19
C (24) SPLIT TO SUBCOOL
      SS24=SS4
C CALCULATE FOR DESIGN HEAT LOAD HEATL(TONS)
C 1 TON = 3.52 KJ/SEC = 211.2 KJ/MIN
C EVMOL  EVAP FLOWRATE MOLES/MIN
      QLOAE= HEATL*211.2
      EVMOL=QLOAE/(H1-H5)
      VRATE=FL1*VVCOM*EVMOL
      VRATE2=FL21*VVCOM2*EVMOL
      EE=EVMOL
      CALL CONAME(IU,ISH,TCOMP)
      CALL UNIT(IU,TCOMP, 1,T1,P1,H1,FL1,YQ1,WM,EE,XMOL,SS1)
      CALL UNIT(IU,TCOMP, 5,T5,P5,H5,FL5,YQ5,WM,EE,XMOL,SS5)
      CALL UNIT(IU,TCOMP, 7,T7,P7,H7,FL7,YQ7,WM,EE,XMOL,SS7)
      CALL UNIT(IU,TCOMP, 4,T4,P4,H4,FL4,YQ4,WM,EE,XMOL,SS4)
      CALL UNIT(IU,TCOMP,24,T4,P4,H4,FL24,YQ4,WM,EE,XMOL,SS24)
      CALL UNIT(IU,TCOMP, 2,T2,P2,H2,FL2,YQ2,WM,EE,XMOL,SS2)
      CALL UNIT(IU,TCOMP,20,T20,P20,H20,FL20,YQ20,WM,EE,XMOL,SS20)
      CALL UNIT(IU,TCOMP,21,T21,P21,H21,FL21,YQ21,WM,EE,XMOL,SS21)
      CALL UNIT(IU,TCOMP,22,T22,P22,H22,FL22,YQ22,WM,EE,XMOL,SS22)
      CALL UNIT(IU,TCOMP,18,T18,P18,H18,FL18,YQ18,WM,EE,XMOL,SS18)
```

```
      CALL UNIT(IU,TCOMP,19,T19,P19,H19,FL19,YQ19,WM,EE,XMOL,SS19)
C     SINK OUTLET TEMPERATURE  (5 F PINCH POINT)
      DTPIN=5./1.8
      IF (YQ22.LE.1) THEN
      T42=T22-DTPIN
      ELSE
      CALL BUBLP(P22,XXX,XV22,T22VS,VL22S,VV22S,.FALSE.,LCRIT)
      CALL HCVCPS(1,T22VS,VV22S,XV22,H22S,CV,CP,VS)
      RAT=(H22-H4)/(H22S-H4)
      T42=(T22VS-DTPIN-T41)*RAT+T41
      ENDIF
      CALL TES(IU,1+DM1+DM2,T41,T22,T4,T42,0.,H22,H4,0.,0.,P22,P4,0.,
     *XMOL,0.,NDIV,EVMOL,UASIC,-1,3)
      Q41=FL22*(H22-H4)
      FL41=Q41/(T42-T41)/CPP(IFSI)
      Q31=(H1-H5)
      FL31=Q31/(T31-T32)/CPP(IFSO)
      CALL SINSOU(FEOUT,FCOUT,IU,FL31,T31,T32,FL41,T41,T42,EE)
      QEVA1=(H1-H5)/WM
      COP= (H1-H5)/ELECP
      IF (IFSI.EQ.1) RHO=101.325/0.287/T41
      POFAN=FL41*DPSI/RHO/FANEF
      ELECT=POFAN+ELECP
      COPT=(H1-H5)/ELECT
      IF (IU.EQ.0) WRITE(6,125)T5,DTHX,QEVA1,VRATE,VRATE2
      IF (IU.EQ.1) WRITE(6,126)T5*1.8-460.,DTHX*1.8,
     $.948*QEVA1/2.20462,VRATE*264.15,VRATE2*264.15
      WRITE(6,127)COP,COPT,PR1,PR2,YQ20
 125  FORMAT(//,30X,'  TEM EVAP IN (K)        =',F10.1,
     &/,        30X,'  DTHX (K)               =',F10.1,
     &/,        30X,'  COOLING RATE (KJ/KG)   =',F10.1,
     &/,        30X,'  SUC1 VOL RATE(M**3/MIN)=',F10.1,
     &/,        30X,'  SUC2 VOL RATE(M**3/MIN)=',F10.1)
 126  FORMAT(//,30X,'  TEM EVAP IN (F)        =',F10.1,
     &/,        30X,'  DTHX (F)               =',F10.1,
     &/,        30X,'  COOLING RATE (BTU/LBM) =',F10.1,
     &/,        30X,'  SUC1 VOL RATE (GPM)    =',F10.1,
     &/,        30X,'  SUC2 VOL RATE (GPM)    =',F10.1)
 127  FORMAT(
     &          30X,'  COP                    =',F10.2,
     &/,        30X,'  COPT                   =',F10.2,
     &/,        30X,'  PRESSURE RATIO 1       =',F10.2,
     &/,        30X,'  PRESSURE RATIO 2       =',F10.2,
     &/,        30X,'  QUALITY M/C OUT        =',F10.2,//)
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C IRREVERSIBILITY ANALYSIS
      WRITE(6,*)'     IRREVERSIBILITY ANALYSIS '
      WRITE(6,*)'      AT TEMPERATURE',TIRR
      WRITE(6,*)
C VALVE 7-5
      CALL FIRRE2(IU,'VALVE 7-5      ',TIRR,FL5,SS7,SS5,WM,EE,0.,0.
     $,TG1)
C VALVE 247-18
      CALL FIRRE2(IU,'VALVE 24-18    ',TIRR,FL18,SS24,SS18,WM,EE,0.,0.
     $,TG2)
C M/C 19-20
```

```
      CALL FIRREC(IU,'M/C 19-20      ',TIRR,.5*(T1+T22),FL19,H19,H20,
     $SS19,SS20,WM,EE,0.,0.,TG3)
C COMP 1-2
      CALL FIRRE2(IU,'COMP 1-2       ',TIRR,FL1,SS1,SS2,WM,EE,0.,
     $0.,TG4)
C COMP 21-22
      CALL FIRRE2(IU,'COMP 21-22     ',TIRR,FL21,SS21,SS22,WM,EE,0.,
     $0.,TG5)
C COND 22-3 & 41-42
      CALL FIRRE4(IU,'COND 22-3,41-42',TIRR,FL22,H22,H4,SS22,SS4,WM,
     *FL41,T41,T42,1.,EE,IFSI,TG6)
C EVAP 5-1 & 31-32
      CALL FIRRE4(IU,'EVAP 5-1,31-32 ',TIRR,FL5,H5,H1,SS5,SS1,WM,
     *FL31,T31,T32,1.,EE,IFSO,TG7)
C SUB1 4-7 & 18-19
      CALL FIRRE4(IU,'SUB1 4-7,18-19 ',TIRR,FL4,H4,H7,SS4,SS7,WM,
     *FL18,SS18,SS19,WM,EE,0,TG8)
C MIX 2-20-21
      CALL FIRRE3(IU,'MIX 2-20-21    ',TIRR,FL2,SS2,WM,FL20,SS20,
     *WM,FL21,SS21,WM,EE,1,TG9)
      TGTOT=TG1+TG2+TG3+TG4+TG5+TG6+TG7+TG8+TG9
C
      WRITE(6,447) TGTOT
 447  FORMAT('*************** TOTAL = ',F10.2,//)
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
      IF (ICHECK.EQ.0) GOTO 130
      IF (IPRI.EQ.1) WRITE(6,*)' HX HIGH 2'
      CALL TES(IU,1+DM1+DM2,T18,T4,T7,T19,H18,H4,H7,H19,P18,P4,P7,P19,
     *XMOL,XMOL,NDIV,EVMOL,UASI2,IPRI,2)
      IF (IPRI.EQ.1) WRITE(6,*)' CONDENSER 3'
      CALL TES(IU,1+DM1+DM2,T41,T22,T4,T42,0.,H22,H4,0.,0.,P22,P4,0.,
     *XMOL,0.,NDIV,EVMOL,UASIC,IPRI,3)
      IF (IPRI.EQ.1) WRITE(6,*)' EVAPORATOR 4'
      CALL TES(IU,0.,T5,T31,T32,T1,H5,0.,0.,H1,P5,0.,0.,P1,0.,
     *XMOL,NDIV,EVMOL,UASIE,IPRI,4)
      UATOTT=UASI2+UASIC+UASIE
      IF (IU.EQ.0) WRITE(90,925)T4,T5,DTHX,DTSUB,COP,COPT,PR1,PR2,
     *FCOUT,FEOUT,UASIC,UASIE,UASI2,UATOTT,VRATE,VRATE2
      IF (IU.EQ.1)  WRITE(90,925)T4*1.8-460.,T5*1.8-460.,DTHX*1.8,
     *DTSUB*1.8,COP,COPT,PR1,PR2,FCOUT,FEOUT,UASIC*.5267,UASIE*.5267,
     *UASI2*.5267,UATOTT*.5267,VRATE*264.15,VRATE2*264.15
 925  FORMAT(1X,4F5.1,4F5.2,F6.1,5F5.1,2F4.0)
 130  CONTINUE
      RETURN
      END
C
CCCCCCCCCCCCCCCCCCCC
```

```
C ABS(ID) =0 SI SYSTEM  ABS(ID)=1  ENGLISH SYSTEM
C ID<0 DATA IS NOT TRANSFERED FOR SCHEMATIC
      IF (ABS(ID).EQ.1) THEN
      UTT=UT*1.8-460.
      UPP=UP*.14504
      UHH=0.948*UH/WW/2.20462
      UFF=UF*EE*WW*2.20462
      USS=0.2389*US/WW
      ELSE
      UTT=UT
      UPP=UP
      UHH=UH/WW
      UFF=UF*EE*WW
      USS=US/WW
      ENDIF
      WRITE(6,20)VNODE,NN,UFF,UPP,UTT,UHH,UY,XNN,WW,USS
      IF (ID.EQ.1) WRITE(75,30)NN,UFF,UPP,UTT,UHH,UY
20    FORMAT (1X,A17,1X,I2,F6.2,F7.1,F7.1,F7.2,F7.2,F7.3,F7.1,F7.3)
30    FORMAT (1X,I9,F9.2,F9.1,F9.1,F9.2,F9.2,F9.1)
      RETURN
      END
C
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C                SUBROUTINE TO CHANGE OUTPUT UNIT
C                  ID= 0 (SI)      ID= 1(ENGLISH)
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
      SUBROUTINE UNITH(ID,QLOAD,PINCH,DTMI,UASI,IIHX,EVMOL)
C DTMI LOGMEAN TEMPERATURE
      IF (PINCH.LT.2.7) UASI=999999.
      IF (ID.EQ.1) THEN
      WRITE(6,11) IIHX,QLOAD*.948*EVMOL,PINCH*1.8,DTMI*1.8,UASI*.5267
      ELSE
      WRITE(6,11) IIHX,QLOAD*EVMOL,PINCH,DTMI,UASI
      ENDIF
 11   FORMAT(
     &30X,'  HX ',I1,' HEAT LOAD   =',F10.0,/,
     &30X,'  PINCH POINT     =',F10.2,/,
     &30X,'  LOGMEAN TEMP.   =',F10.2,/,
     &30X,'  UA              =',F10.2)
      RETURN
      END
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C               SUBROUTINE TO CALL SUBROUTINE CHECKE,CHEKC,CHECK
C  TO CHECK THE PINCH POINT OF EVAPORATOR,CONDENSER AND SUBCOOLER
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
      SUBROUTINE TES(IU,DMHF,TCRI,THRI,THRO,TCRO,HCRI,HHRI,HHRO,HCRO,
     *PCRI,PHRI,PHRO,PCRO,XMH,XMC,NDIV,EVMOL,UASI,IPRI,IIHX)
C     IU    : UNIT SYSTEM
```

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C     DMHF  : HOT SIDE FLOWRATE (KMOLE/MIN)
C     TCRI  : INLET COLD SIDE TEMPERATURE (C)
C     THRI  : INLET HOT SIDE TEMPERATURE (C)
C     TCRO  : OUTLET COLD SIDE TEMPERATURE (C)
C     THRI  : OUTLET HOT SIDE TEMPERATURE (C)
C     HCRI  : INLET COLD SIDE ENTHALPY KJ/KMOLE
C     HHRI  : INLET HOT SIDE ENTHALPY KJ/KMOLE
C     HCRO  : OUTLET COLD SIDE ENTHALPY KJ/KMOLE
C     HHRO  : OUTLET HOT SIDE ENTHALPY KJ/KMOLE
C     PCRI  : INLET COLD SIDE PRESSURE KPA
C     PHRI  : INLET HOT SIDE PRESSURE KPA
C     PCRO  : OUTLET COLD SIDE PRESSURE KPA
C     PHRO  : OUTLET HOT SIDE PRESSURE KPA
C     XMC   : MOLE FRACTION OF FIRST REFRIGERANT COLD SIDE
C     XMH   : MOLE FRACTION OF FIRST REFRIGERANT HOT  SIDE
C     NDIV  : NUMBER OF SEGMENTS
C     EVMOL : MOLE FLOW RATE OF EVAPORATOR FLOW KMOLE/MIN
C     UASI  : CONDUCTANCE
C     IIHX  : INDICATOR FOR SUBCOOLER, CONDENSER OR EVAPORATOR
      DPH=(PHRI-PHRO)/NDIV
      DPC=(PCRI-PCRO)/NDIV
      IF (HCRI.EQ.0.) GOTO 501
      IF (HHRI.EQ.0.) GOTO 502
C SUBCOOLER
      CALL CHECK(IU,TCRI,THRI,THRO,TCRO,HCRI,HHRI,HHRO,HCRO,PHRO,
     *PCRI,DPH,DPC,XMH,XMC,DTMI,NDIV,PINCH,IPRI)
      QLOAD=DMHF*(HHRI-HHRO)
      GOTO 505
C CONDENSER
 501  CONTINUE
      IF (IPRI.EQ.-1) THEN
      TCO=TCRO
      TCOMI=TCO-2.
      TCOMA=TCO+2.
      ITERC=0
 642  CALL CHECKC(IU,TCRI,THRI,THRO,TCO,HHRI,HHRO,PHRO,DPH,XMH,
     *DTMI,NDIV,PINCH,0)
      IF (ABS(PINCH/2.77778-1).LE.0.01) GOTO 643
      CALL CNVRG(TCO,TCOMI,TCOMA,PINCH,2.7778,ITERC,DUM1,DUM2,6)
      IF (ITERC-10) 642,642,644
  644 WRITE(6,*) ' TROUBLE 644'
      RETURN
 643  TCRO=TCO
      RETURN
      ENDIF
      CALL CHECKC(IU,TCRI,THRI,THRO,TCRO,HHRI,HHRO,PHRO,DPH,XMH,
     *DTMI,NDIV,PINCH,IPRI)
      QLOAD=DMHF*(HHRI-HHRO)
      GOTO 505
C EVAPORATOR
 502  CALL CHECKE(IU,TCRI,THRI,THRO,TCRO,HCRI,HCRO,PCRI,
     *DPC,XMC,DTMI,NDIV,PINCH,IPRI)
      QLOAD=(HCRO-HCRI)
C     KJ/K
  505 UASI=QLOAD/DTMI*EVMOL
      CALL UNITH(IU,QLOAD,PINCH,DTMI,UASI,IIHX,EVMOL)
```

```
      RETURN
      END
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C                SUBROUTINE TO CHECK SUBCOOLER
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
      SUBROUTINE CHECK(IU,TLST,THEND,THST,TLEND,ELST,EHEND,EHST,ELEND,
     1       PH,PL,DP1,DP2,XMOLH,XMOLL,DTMI,NDIV,PINCH,IPRI)
      IMPLICIT REAL*8 (A-H,O-Z)
      DIMENSION ELA(200),EHA(200),DELT(200),TLA(200),THA(200)
C
C THE FUNCTION OF THIS SUBROUTINE IS TO DETERMINE IF HX1 OR HX2
C VIOLATES THE 2ND LAW OF THERMODYNAMICS
C
      NDIV1=NDIV+1
      DO 50 I = 1,NDIV1
         THA(I) = 0.
         TLA(I) = 0.
 50   CONTINUE
      DH1=(EHEND-EHST)/NDIV
      DH2=(ELEND-ELST)/NDIV
C
      EHA(1) = EHST
      ELA(1) = ELST
      THA(1) = THST
      TLA(1) = TLST
      DELT(1)  = THST-TLST
      IFLAG  = 0
C
      DO 100 I =1,NDIV
         AI = I-1
         EHA(I) = EHST+AI*DH1
         ELA(I) = ELST+AI*DH2
         PTH=PH+AI*DP1
         PTL=PL-AI*DP2
         CALL HPIN(EHA(I),PTH,XMOLH,THA(I),YQ,XL,XV,VL,VV,HL,HV)
         CALL HPIN(ELA(I),PTL,XMOLL,TLA(I),YQ,XL,XV,VL,VV,HL,HV)
         DELT(I)  = THA(I) - TLA(I)
          IF (DELT(I) .LT. 0.) IFLAG = 1
         J = I+1
 100  CONTINUE
      EHA(J) = EHEND
      ELA(J) = ELEND
      THA(J) = THEND
      TLA(J) = TLEND
      DELT(J)  = THEND-TLEND
           IF (IFLAG .EQ. 0) THEN
           SDTMI  = 0.
             DO 200 I = 1,NDIV
           DT1  = THA(I+1)-TLA(I+1)
           DT2  = THA(I)-TLA(I)
           IF (DT1 .EQ. DT2) THEN
             DTMI = DT1
```

```
           ELSE
             DTM  = (THA(I+1)-TLA(I+1)) - (THA(I)-TLA(I))
             DTML = LOG((THA(I+1)-TLA(I+1))/(THA(I)-TLA(I)))
             DTMI = DTM/DTML
           ENDIF
          SDTMI=SDTMI+DTMI
          DTMI=SDTMI/NDIV
200     CONTINUE
           ELSE
        DTMI= 0.
       ENDIF
C
 6410 FORMAT(5X,I5,5F10.2)
      PINCH=2000.
      DO 300 I=1,NDIV1
      PIT=THA(I)-TLA(I)
      IF (PIT.LT.PINCH) PINCH=PIT
      IF (IPRI.EQ.1) THEN
      IF (IU.EQ.0) WRITE(6,6410) I,EHA(I),ELA(I),THA(I),TLA(I),DELT(I)
      IF (IU.EQ.1) WRITE(6,6410) I,THA(I)*1.8-460.,TLA(I)*1.8-460.
     *,DELT(I)*1.8
      ENDIF
 300  CONTINUE
      RETURN
      END
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C                SUBROUTINE TO CHECK CONDENSER
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
      SUBROUTINE CHECKC(IU,TLST,THEND,THST,TLEND,EHEND,EHST,
     1       PH,DP1,XMOLH,DTMI,NDIV,PINCH,IPRI)
      IMPLICIT REAL*8 (A-H,O-Z)
      DIMENSION EHA(200),DELT(200),TLA(200),THA(200)
C
C CONDENSER
C
      NDIV1=NDIV+1
      DO 50 I = 1,NDIV1
         THA(I) = 0.
         TLA(I) = 0.
 50   CONTINUE
C
      DH1=(EHEND-EHST)/NDIV
      EHA(1) = EHST
      THA(1) = THST
      TLA(1) = TLST
      DELT(1)  = THST-TLST
      IFLAG  = 0
      DTTE=(TLEND-TLST)/NDIV
      DO 100 I =1,NDIV
         AI = I-1
         EHA(I) = EHST+AI*DH1
         TLA(I) = TLST+AI*DTTE
```

```
         PTH=PH+AI*DP1
         CALL HPIN(EHA(I),PTH,XMOLH,THA(I),YQ,XL,XV,VL,VV,HL,HV)
         DELT(I)  = THA(I) - TLA(I)
         IF (DELT(I) .LT. 0.) IFLAG = 1
         J = I+1
 100  CONTINUE
      EHA(J) = EHEND
      THA(J) = THEND
      TLA(J) = TLEND
      DELT(J)  = THEND-TLEND
C
      IF (IFLAG .EQ. 0) THEN
         SDTMI  = 0.
         DO 200 I = 1,NDIV
           DT1  = THA(I+1)-TLA(I+1)
           DT2  = THA(I)-TLA(I)
           IF (DT1 .EQ. DT2) THEN
             DTMI = DT1
           ELSE
             DTM  = (THA(I+1)-TLA(I+1)) - (THA(I)-TLA(I))
             DTML = LOG((THA(I+1)-TLA(I+1))/(THA(I)-TLA(I)))
             DTMI = DTM/DTML
           ENDIF
        SDTMI=SDTMI+DTMI
         DTMI=SDTMI/NDIV
 200    CONTINUE
       ELSE
        DTMI= 0.
      ENDIF
C
 6410 FORMAT(5X,I5,4F10.2)
      PINCH=2000.
      DO 300 I=1,NDIV1
      PIT=THA(I)-TLA(I)
      IF (PIT.LT.PINCH) PINCH=PIT
      IF (IPRI.EQ.1) THEN
      IF (IU.EQ.0) WRITE(6,6410) I,EHA(I),THA(I),TLA(I),DELT(I)
      IF (IU.EQ.1) WRITE(6,6410) I,THA(I)*1.8-460.,TLA(I)*1.8-460.
     %,DELT(I)*1.8
      ENDIF
 300  CONTINUE
      RETURN
      END
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C                SUBROUTINE TO CHECK EVAPORATOR
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
      SUBROUTINE CHECKE(IU,TLST,THEND,THST,TLEND,ELST,ELEND,
     1      PL,DP2,XMOLL,DTMI,NDIV,PINCH,IPRI)
      IMPLICIT REAL*8 (A-H,O-Z)
      DIMENSION ELA(200),DELT(200),TLA(200),THA(200)
C
C THE FUNCTION OF THIS SUBROUTINE IS TO DETERMINE THE PINCH POINT
C AND CONDUCTANCE OF EVAPORATOR
```

```
C
      NDIV1=NDIV+1
      DO 50 I = 1,NDIV1
         THA(I) = 0.
         TLA(I) = 0.
 50   CONTINUE
      DH2=(ELEND-ELST)/NDIV
      DTTE=(THEND-THST)/NDIV
C
      ELA(1) = ELST
      THA(1) = THST
      TLA(1) = TLST
      DELT(1)  = THST-TLST
      IFLAG  = 0
C
      DO 100 I =1,NDIV
         AI = I-1
         THA(I) = THST+AI*DTTE
         ELA(I) = ELST+AI*DH2
         PTL=PL-AI*DP2
 888  FORMAT(6F12.3)
      IF (XMOLL.LE.0.001.OR.XMOLL.GE.0.999) THEN
      CALL BUBLP(PTL,XML,XMOLL,TLA(I),VLL,VVV,.FALSE.,LCRIT)
      ELSE
      CALL HPIN(ELA(I),PTL,XMOLL,TLA(I),YQ,XL,XV,VL,VV,HL,HV)
      ENDIF
C     WRITE(6,888) ELA(I),PTL,XMOLL,TLA(I)
      DELT(I)  = THA(I) - TLA(I)
          IF (DELT(I) .LT. 0.) IFLAG = 1
         J = I+1
 100  CONTINUE
      ELA(J) = ELEND
      THA(J) = THEND
      TLA(J) = TLEND
      DELT(J)  = THEND-TLEND
           IF (IFLAG .EQ. 0) THEN
            SDTMI  = 0.
              DO 200 I = 1,NDIV
           DT1  = THA(I+1)-TLA(I+1)
           DT2  = THA(I)-TLA(I)
           IF (DT1 .EQ. DT2) THEN
             DTMI = DT1
           ELSE
             DTA= (THA(I+1)-TLA(I+1)) + (THA(I)-TLA(I))/2.
             DTM = (THA(I+1)-TLA(I+1)) - (THA(I)-TLA(I))
             DDDD=(THA(I+1)-TLA(I+1))/(THA(I)-TLA(I))
             IF (DDDD.LE.0.) THEN
             DTML=0.00001
             ELSE
             DTML = LOG(DDDD)
             ENDIF
             DTMI = DTM/DTML
           ENDIF
        SDTMI=SDTMI+DTMI
         DTMI=SDTMI/NDIV
 200    CONTINUE
```

```
          ELSE
        DTMI= 0.
       ENDIF
C
 6410 FORMAT(5X,I5,4F10.2)
      PINCH=2000.
      DO 300 I=1,NDIV1
      PIT=THA(I)-TLA(I)
      IF (PIT.LT.PINCH) PINCH=PIT
      IF (IPRI.EQ.1)THEN
      IF (IU.EQ.0) WRITE(6,6410) I,ELA(I),THA(I),TLA(I),DELT(I)
      IF (IU.EQ.1) WRITE(6,6410) I,THA(I)*1.8-460.,TLA(I)*1.8-460.
     &,DELT(I)*1.8
      ENDIF
 300  CONTINUE
C     WRITE(6,*)' DTA =',DTA
      RETURN
      END
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C             SUBROUTINE  OUTPUT
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
      SUBROUTINE CONAME(IU,ISH,T)
      CHARACTER *17 T(50)
      WRITE(6,*)'      NODE           FLOW    PRESS.  TEMP.  ENTHAL.
     *     QUA      WM   ENTROPY'
      IF (IU.EQ.1) THEN
      WRITE(6,*)'                    LB/MIN   PSIA    F     BTU/LBM
     *     KG/KMOL  BTU/LBM-R'
      ELSE
      WRITE(6,*)'                    KG/MIN   KPA   K       KJ/KG
     *     KG/KMOL  KJ/KG-K'
      ENDIF
      T(1)='EVAPORATOR OUT '
      T(4)='SUB HIGH HOT IN'
      T(5)='EVAPORATOR IN'
      T(6)='SUB LOW HOT OUT'
      T(7)='SUB LOW HOT IN'
      T(25)='SUB LOW COLD IN'
      T(26)='SUB LOW COLD OUT'
      T(31)='SOURCE IN'
      T(32)='SOURCE OUT'
      T(41)='SINK IN'
      T(42)='SINK OUT'
      IF (ISH.EQ.0) GOTO 100
      IF (ISH.EQ.1) GOTO 110
      IF (ISH.EQ.2) GOTO 120
      IF (ISH.EQ.3) GOTO 130
      IF (ISH.EQ.4) GOTO 140
      IF (ISH.EQ.5) GOTO 150
      IF (ISH.EQ.6) GOTO 160
      GOTO 500
 100  CONTINUE
```

```
C  PURE 2 STAGE COMPRESSOR
      T(7)='SUB.  HOT OUT'
      T(4)='SUB. HOT IN'
      T(21)='COMPRESSOR II IN'
      T(22)='COMPRESSOR II OUT'
      T(2)='COMPRESSOR I OUT'
      T(18)='SUB. COLD IN'
      T(19)='SUB. COLD OUT'
      T(20)='M/C OUT'
      T(24)='SLIT TO SUB I'
      GOTO 500
 110  CONTINUE
C SUNDSTRAND
      T(21)='COMPRESSOR II IN'
      T(22)='COMPRESSOR II OUT'
      T(11)='COMPRESSOR I IN'
      T(17)='TXV1 IN'
      T(2)='COMPRESSOR I OUT'
      T(18)='SUB HIGH COLD IN'
      T(19)='SUB HIGH COLD OUT'
      T(20)='M/C OUT'
      T(27)='SUB HIGH OUT'
      GOTO 500
 120  CONTINUE
C NOVEL WITH LIQUID PUMP
      T(21)='COMPRESSOR IN'
      T(22)='COMPRESSOR OUT'
      T(2)='SUB HIGH COLD IN'
      T(8)='SUB HIGH COLD OUT'
      T(10)='PUMP IN'
      T(9)='PUMP OUT'
      T(14)='M/C OUT'
      T(23)='CONDENSER IN'
      GOTO 500
 130  CONTINUE
C NOVEL WITH LIQUID PUMP LIQUID RETURN TO OUTLET CONDENSER
      T(21)='COMPRESSOR IN'
      T(22)='COMPRESSOR OUT'
      T(2)='SUB HIGH COLD IN'
      T(8)='SUB HIGH COLD OUT'
      T(10)='PUMP IN'
      T(9)='PUMP OUT'
      T(14)='M/C OUT'
      T(3)='CONDENSER OUT'
      GOTO 500
 140  CONTINUE
C NOVEL WITH LIQUID PUMP AND HX III
      T(21)='COMPRESSOR IN'
      T(22)='COMPRESSOR OUT'
      T(3) ='CONDENSER OUT'
      T(2)='SUB HIGH COLD IN'
      T(8)='SUB HIGH COLD OUT'
      T(20)='SUB III HOT IN'
      T(10)='PUMP IN'
      T(9)='PUMP OUT'
      T(14)='M/C OUT'
```

```
      T(15)='SUB. III COLD OUT'
      GOTO 500
 150  CONTINUE
C NOVEL WITHOUT LIQUID PUMP
      T(21)='COMPRESSOR IN'
      T(22)='COMPRESSOR OUT'
      T(2)='SUB HIGH COLD IN'
      T(8)='SUB HIGH COLD OUT'
      GOTO 500
 160  CONTINUE
C LOZEN
      T(21)='COMPRESSOR IN'
      T(22)='COMPRESSOR OUT'
      T(20)='M/C OUT       '
      T(19)='M/C IN'
      T(4) ='CONDENSER OUT'
      GOTO 500
 500  CONTINUE
      RETURN
      END
C
      SUBROUTINE SINSOU(FFSI,FFSO,IU,FLOI,TSOI,TSOO,FLII,TSII,TSIO,EMOL)
      DIMENSION TW(4)
      TW(1)=TSOI
      TW(2)=TSOO
      TW(3)=TSII
      TW(4)=TSIO
C     IU=0  SI SYSTEM
C     IU=1  ENGLISH
      IF (IU.EQ.1) THEN
      FFSI=FLII*EMOL*2.2046
      FFSO=FLOI*EMOL*2.2046
      ELSE
      FFSI=FLII*EMOL
      FFSO=FLOI*EMOL
      ENDIF
      IF (IU.EQ.1) THEN
      DO 10 J=1,4
 10   TW(J)=TW(J)*1.8-460.
      ENDIF
      WRITE(75,20) (TW(K),K=1,4)
      WRITE(6,30)  FFSO,(TW(K),K=1,2),FFSI,(TW(K),K=3,4)
 20   FORMAT(4F10.2)
 30   FORMAT(' SOURCE IN     ',6X,F6.2,8X,F6.1,/,
     *       ' SOURCE OUT    ',20X,F6.1,/,
     *       ' SINK   IN     ',6X,F6.2,8X,F6.1,/,
     *       ' SINK   OUT    ',20X,F6.1,/)
      RETURN
      END
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C    IRREVERSIBILITY ANALYSIS
C
C    IRREVERSIBILITY ANALYSIS
C    FIRE2     2 LEGS COMPONENT
      SUBROUTINE FIRRE2(ID,TITC,UTO,UF,USI,USO,WW,EE,UQ,UW,ZIRRT)
      CHARACTER *15 TITC
```

```
      IF (ABS(ID).EQ.1) THEN
      UTTO=UTO*1.8
      UFF=UF*EE*WW*2.20462
      USSI=0.2389*USI/WW
      USSO=0.2389*USO/WW
      UQQ=UQ
      UWW=UW
      ELSE
      UTTO=UTO
      UFF=UF*EE*WW
      USSI=USI/WW
      USSO=USO/WW
      UQQ=UQ
      UWW=UW
      ENDIF
      ZIRRT=UFF*UTTO*(USSO-USSI)
      WRITE(6,10)TITC,UFF,USSI,USSO,UQQ,UWW,WW,ZIRRT
   10 FORMAT(1X,A15,2X,7F8.3)
      RETURN
      END
C     FIRREC     MOTOR COOLING
      SUBROUTINE FIRREC(ID,TITC,UTO,UTA,UF,UHI,UHO,USI,USO,WW,EE,UQ,
     &UW,ZIRRT)
      CHARACTER *15 TITC
      IF (ABS(ID).EQ.1) THEN
      UTTO=UTO*1.8
      UTTA=UTA*1.8
      UFF=UF*EE*WW*2.20462
      USSI=0.2389*USI/WW
      USSO=0.2389*USO/WW
      UHHI=0.948*UHI/WW/2.20462
      UHHO=0.948*UHO/WW/2.20462
      UQQ=UQ
      UWW=UW
      ELSE
      UTTO=UTO
      UTTA=UTA
      UFF=UF*EE*WW
      USSI=USI/WW
      USSO=USO/WW
      UHHI=UHI/WW
      UHHO=UHO/WW
      UQQ=UQ
      UWW=UW
      ENDIF
      ZIRRT=UFF*UTTO*((USSO-USSI)+(UHHI-UHHO)/UTTA)
      WRITE(6,10)TITC,UFF,USSI,USSO,UHHI,UHHO,WW,ZIRRT,UTTA
   10 FORMAT(1X,A15,2X,5F7.4,F7.2,F7.3,F7.1)
      RETURN
      END
C FIRE4       4 LEGS COMPONENT
      SUBROUTINE FIRRE4(ID,TITC,UTO,UF1,UHI1,UHO1,USI1,USO1,WW1,
     *UF2,DUMI2,DUMO2,WW2,EE,IFLUID,ZIRRT)
      CHARACTER *15 TITC
C     REF   USI2,USO2  INLET OUTLET ENTROPY
C     AIR,WATER USI2,USO2  INLET,OUTLET TEMPERATURE
```

```
C SPECIFIC HEAT  AIR IFLUID=1.(1. KJ/KG-C), WATER IFLUID=2 (4.18 KJ/KG-C
      IF (IFLUID.EQ.0) THEN
      CP=0.
      USI2=DUMI2
      USO2=DUMO2
      ENDIF
      IF (IFLUID.EQ.1) THEN
      CP=1.
      TCI=DUMI2
      TCO=DUMO2
      USI2=1.4143+CP*LOG(TCI/100.)
      USO2=1.4143+CP*LOG(TCO/100.)
      DELT=TCI-TCO
      ENDIF
C     IF (IFLUID.EQ.2) CP=4.18
      IF (ID.EQ.1) THEN
      CPP=CP*0.2388
      DELTT=DELT*1.8
      UTTO=UTO*1.8
      UFF1=UF1*EE*WW1*2.20462
      UFF2=UF2*EE*WW2*2.20462
      UHHI1=0.948*UHI1/WW1/2.20462
      UHHO1=0.948*UHO1/WW1/2.20462
      USSI1=0.2389*USI1/WW1
      USSI2=0.2389*USI2/WW2
      USSO1=0.2389*USO1/WW1
      USSO2=0.2389*USO2/WW2
      ELSE
      CPP=CP
      DELTT=DELT
      UTTO=UTO
      UFF1=UF1*EE*WW1
      UFF2=UF2*EE*WW2
      UHHI1=UHI1/WW1
      UHHO1=UHO1/WW1
      USSI1=USI1/WW1
      USSI2=USI2/WW2
      USSO1=USO1/WW1
      USSO2=USO2/WW2
      ENDIF
      ZIRR1=UFF1*UTTO*(USSO1-USSI1)
      IF (IFLUID.EQ.0) GOTO 5
C AIR SOURCE SINK
      UFF2=ABS(UFF1*(UHHI1-UHHO1)/CPP/DELTT)
C
  5   ZIRR2=UFF2*UTTO*(USSO2-USSI2)
      ZIRRT=ZIRR1+ZIRR2
      WRITE(6,10)TITC,UFF1,USSI1,USSO1,WW1
      WRITE(6,11)          UFF2,USSI2,USSO2,ZIRRT
  10  FORMAT(1X,A15,2X,3F8.3,16X,F8.3)
  11  FORMAT(1X,17X,3F8.3,24X,3F8.3)
      RETURN
      END
C FIRRE3       3 LEGS COMPONENT
      SUBROUTINE FIRRE3(ID,TITC,UTO,UF1,US1,WW1,UF2,US2,WW2
     *,UF3,US3,WW3,EE,IMIX,ZIRRT)
```

```
      CHARACTER *15 TITC
C     IMIX=1 MIXING
C     IMIX=-1 SEPARATOR
      IF (ID.EQ.1) THEN
      UTTO=UTO*1.8
      UFF1=UF1*EE*WW1*2.20462
      UFF2=UF2*EE*WW2*2.20462
      UFF3=UF3*EE*WW3*2.20462
      USS1=0.2389*US1/WW1
      USS2=0.2389*US2/WW2
      USS3=0.2389*US3/WW3
      ELSE
      UTTO=UTO
      UFF1=UF1*EE*WW1
      UFF2=UF2*EE*WW2
      UFF3=UF3*EE*WW3
      USS1=US1/WW1
      USS2=US2/WW2
      USS3=US3/WW3
      ENDIF
      ZIRR1=UFF1*UTTO*USS1
      ZIRR2=UFF2*UTTO*USS2
      ZIRR3=UFF3*UTTO*USS3
      IF (IMIX.EQ.1) THEN
      ZIRRT=ZIRR3-ZIRR1-ZIRR2
      ELSE
      ZIRRT=ZIRR3+ZIRR2-ZIRR1
      ENDIF
      WRITE(6,10)TITC,UFF1,USS1,WW1
      WRITE(6,11)     UFF2,USS2,WW2
      WRITE(6,11)     UFF3,USS3,WW3,ZIRRT
   10 FORMAT(1X,A15,2X,2F8.3,24X,F8.3)
   11 FORMAT(1X,17X,2F8.3,24X,2F8.3)
      RETURN
      END
```

VITA

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Doctor of Philosophy

Thesis: NON-AZEOTROPIC REFRIGERANT MIXTURE (NARM) VAPOR COMPRESSION CYCLES

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